TECHNICAL FEASIBILITY OF REDUCING NO_x AND PARTICULATE EMISSIONS FROM HEAVY-DUTY ENGINES

FINAL REPORT

April 30, 1993

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Prepared For

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- Diesel control technologies (Section 3)
- Gaseous fuel technologies (Section 6)
- Research needs (Section 9)
- Conclusions (Section 10) as they relate to the above sections

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DISCUSSION

As shown within this report, low emissions can be obtained from heavy-duty engines. These emission levels are already achieved by current technology methanol or natural gas. For diesel-fueled engines to meet these levels, new or radically redesigned versions of current engines and a significant advancement in aftertreatment technology will be required. Heavy-duty gasoline engines will require a breakthrough in high temperature three-way catalysts. Tables 1-5 through 1-8 show projected engine and aftertreatment technologies availabilities and emission levels in each classification for light heavy-duty truck engines, medium heavy-duty truck engines, heavy heavy-duty truck engines and urban transit bus engines, respectively.

There are several interesting research efforts that will thrust diesel engines below the 4 g/bhp-hr NO_x set by the EPA for 1998. By using a combination of very high pressure fuel injection, variable geometry turbocharger, enhanced air-to-air aftercooler, optimized combustion chamber, electronic unit injectors with minimized sac volumes, rate shaping, exhaust gas recirculation and sophisticated electronic control of all engine systems, diesel engines could meet a 2.5 g/bhp-hr NO_x standard at 0.15 g/bhp-hr PM with a 5 percent penalty in fuel economy. Advanced oxidation catalysts might reduce PM emissions to less than 0.1 g/bhp-hr while particulate traps could be used to reduce the particulate emissions to 0.05 g/bhp-hr. Durability of the engine may be reduced to 80 percent of the 1994 counterpart. Several manufacturers and research organizations are studying these refinements in diesel technology and predict such engines may be available as early as 2000. Concentrated research and development, tied with demonstration programs, will be needed to bring these engines into reality.

Another interesting effort for reducing diesel NO_x emissions is in the area of catalytic aftertreatment. Two types of catalysts show promise for reducing diesel exhaust emissions. The first is an advanced oxidation catalyst that will reduce PM emissions 20 to 40 percent (depending upon particulate composition) and gaseous HC emission 50 to 60 percent. These will require additional breakthroughs in noble metals and catalyst washcoating. However, advanced oxidation

Table 1-5. Projected light heavy-duty technologies

Projected Fuel/Vehicle Systems	NO_x (g/bhp-hr)	PM (g/bhp-hr)	Year[a] Available
Gasoline Engine with Advanced TWC	3	0.10	1999
Gasoline Engine with Electrically Heated TWC Stoichiometric Methanol Engine with TWC Stoichiometric CNG/LNG Engine with TWC Stoichiometric LPG Engine with TWC	2	0.05	2002 1996 1996 1996
Stoichiometric Methanol Engine with TWC Stoichiometric CNG/LNG Engine with TWC Stoichiometric LPG Engine with TWC	1	0.05	1998 1998 1998
Battery Powered vehicle	0	0	1998

[a] Pre-production field test units

Table 1-6. Projected medium heavy-duty technologies

Projected Fuel/Vehicle Systems	NO_x (g/bhp-hr)	PM (g/bhp-hr)	Year[a] Available
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and oxidation catalyst	3	0.10	2000 1999
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and DE-NO_x & oxidation cat. DI diesel with EGR and catalytic trap 2-Stroke DI methanol engine with oxidation catalyst 4-Stroke DI methanol engine with oxidation catalyst Stoichiometric methanol engine with TWC Lean burn CNG/LNG engine with oxidation catalyst Stoichiometric CNG/LNG engine with TWC Stoichiometric LPG engine with TWC	2	0.05	2002 2002 2002 1994 1997 1997 1994 1994 1994
2-Stroke DI methanol engine with oxidation catalyst Stoichiometric methanol engine with TWC Lean burn CNG/LNG engine with oxidation catalyst Lean burn LPG engine with oxidation catalyst Stoichiometric CNG/LNG engine with TWC Stoichiometric LPG engine with TWC Hybrid/Electric truck	1	0.05	1998 2001 2001 2001 1998 1998 2002
Battery Powered vehicle Fuel Cell vehicle	0	0	1998 2000

[a] Pre-production field test units

Table 1-7. Projected heavy heavy-duty technologies

Projected Fuel/Vehicle Systems	NO_x (g/bhp-hr)	PM (g/bhp-hr)	Year[a] Available
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and oxidation catalyst	3	0.10	2000 1999
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and catalytic trap DI diesel with EGR and DE-NO_x & oxidation cat. 2-Stroke DI methanol engine with oxidation catalyst 4-Stroke DI methanol engine with oxidation catalyst Lean burn CNG/LNG engine with oxidation catalyst Lean burn LPG engine with oxidation catalyst	2	0.05	2002 2002 2002 1996 1997 1997 1997
2-Stroke DI methanol engine with oxidation catalyst Lean burn CNG/LNG engine with oxidation catalyst Lean burn LPG engine with oxidation catalyst Hybrid/Electric truck	1	0.05	2001 2001 2001 2002

[a] Pre-production field test units available

Table 1-8. Projected urban transit bus technologies

Projected Fuel/Vehicle Systems	NO_x (g/bhp-hr)	PM (g/bhp-hr)	Year[a] Available
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and oxidation catalyst DI diesel with EGR and catalytic trap	3	0.10	2000 1999 1999
DI diesel with DE-NO_x and oxidation catalyst DI diesel with EGR and DE-NO_x & oxidation cat. 2-Stroke DI methanol engine with oxidation catalyst Stoichiometric methanol engine with TWC Lean burn CNG/LNG engine with oxidation catalyst Lean burn LPG engine with oxidation catalyst Stoichiometric CNG/LNG engine with TWC Stoichiometric LPG engine with TWC	2	0.05	2002 2002 1992 1994 1992 1994 1992 1994
2-Stroke DI methanol engine with oxidation catalyst Stoichiometric methanol engine with TWC Lean burn CNG/LNG engine with oxidation catalyst Lean burn LPG engine with oxidation catalyst Stoichiometric CNG/LNG engine with TWC Stoichiometric LPG engine with TWC Hybrid/Electric bus	1	0.05	1998 1998 1998 1998 1998 1998 1998
Battery Powered bus Fuel Cell bus	0	0	1996 1996

[a] Pre-production field test units available

catalysts can complement new engine technology by allowing manufacturers to concern themselves with NO_x control and use advanced exhaust aftertreatment to reduce excess particulates. Engine manufacturers may want to use this technique for their light and medium heavy-duty engines to minimize development and add-on costs. It is expected that there should be very little degradation in fuel economy using an oxidation catalyst, but that very low sulfur (less than 0.01 percent) fuels will need to be used. These catalysts are estimated to be available in the next 3 to 5 years.

The second type of catalytic aftertreatment is the lean NO_x catalyst. Lean NO_x catalysts use copper zeolites and a reductant to remove NO_x emissions in a fuel-lean environment. There are two types of lean NO_x catalysts currently being developed. The first of these is the selective catalytic reduction (SCR) system which uses ammonia or urea as a reductant. The reductant must be stored on the vehicle. For every gram of NO_x, 0.6 grams of ammonia must be consumed. With this ratio, an 80 percent reduction in NO_x emissions can be realized. Further work to simplify, miniaturize and control the system is needed.

The second type of lean NO_x catalyst is the DE-NO_x (Diesel Engine NO_x) catalyst, which uses copper zeolite sieves to capture exhaust hydrocarbons during idle and low load operation. These trapped hydrocarbons are then used to reduce NO_x emissions during high load operation. Presently DE-NO_x catalysts are only 10 to 20 percent effective in laboratory settings under steady state conditions, yet by decreasing the air/fuel ratio from over 22 to 16 or 18, researchers believe that exhaust temperatures and hydrocarbon concentrations will be enough to increase catalyst NO_x reduction efficiencies to 50 to 70 percent. However, in order to be effective in real world environments, research must find a way for DE-NO_x catalysts to work over the range of temperatures found in diesel engines, the variety of hydrocarbons present and the very lean air/fuel ratios at which diesel engines operate. Navistar claims that their hydraulically-actuated electronic unit injector (HEUI) system is capable of operating at air/fuel ratios of 16 or 18 without significant increases in smoke emissions, but overall particulate emissions are expected to increase. Fuel consumption is estimated to increase 10 to 20 percent at those lower air/fuel ratios, but further development in fuel injection system and catalyst design can limit the fuel economy degradation. Such a system could be available for testing by 1998.

Clean diesel fuel also will play an important role in future diesel engines. With the addition of cetane improvers and oxygenates, CO, HC and PM emissions can be reduced substantially. Additional fuel regulations will be needed to further reduce PM emissions.

Gasoline heavy-duty engines currently are producing 3 to 3.5 g/bhp-hr NO_x. With cleaner gasoline, additional EGR, and significant breakthroughs in high temperature three-way catalyst materials, gasoline engines could meet 2 g/bhp-hr NO_x. Higher temperature three-way catalysts are needed to handle the range of exhaust temperatures typical in heavy-duty gasoline engines. Such catalysts could be available by 1999.

Alternative fueled engines already produce low emissions in heavy-duty engines. Two engines are currently certified and several others are in the process of certification. When discussing alternative fuels and relative fuel economy, it is convenient to refer to diesel equivalent fuel economy. This refers to the energy equivalent of a given fuel to a gallon of diesel fuel. For instance, methanol has an energy density of 57,000 Btu per gallon while diesel fuel has an energy density of 128,000 Btu per gallon. This means it takes 2.25 gallons of methanol to equal one gallon of diesel fuel on an energy basis. Thus a methanol vehicle getting 10 miles per gallon would have a diesel equivalent mileage of 22.5 miles per gallon. This terminology provides a convenient comparison when dealing with alternative fuels of varying energy densities and storage media.

Alcohol fuels already produce low emissions in heavy-duty engines. Detroit Diesel Corporation has certified their 2-stroke direct-injection 6V-92TA engine on methanol at 1.7 g/bhp-hr NO_x and 0.03 g/bhp-hr PM. DDC is the only manufacturer who has continued to develop a methanol engine past the demonstration stage. With better air management, an improved fuel injection system, better oil control and further optimization for methanol use, this engine could reach 1 g/bhp-hr NO_x and still keep particulates around 0.05 g/bhp-hr. Such an engine would cost approximately the same as an advanced diesel engine with EGR and catalytic particulate trap.

Diesel equivalent fuel economy is about 5 to 10 percent lower than the diesel baseline engine. While durability might be somewhat reduced, it is expected that the life cycle costs will only be 10 percent more than the 1998 diesel baseline engine.

Four-stroke direct-injection methanol-fueled engines also show promise as being low emission engines. Currently there are no four-stroke direct-injection engines certified, however, research and development continues on these engines. Several manufacturers are investigating the use of EGR to further lower NO_x emissions, making it a candidate for meeting 1 g/bhp-hr NO_x. Stoichiometric three-way catalyst (TWC) methanol engines can easily meet 1 g/bhp-hr NO_x for light and medium-duty requirements.

Natural gas shows considerable promise as a heavy-duty engine fuel. Cummins recently certified its lean-burn homogeneous-charge L10 engine at 2 g/bhp-hr NO_x and 0.02 g/bhp-hr PM. Homogeneous stoichiometric engines with three-way catalysts are being demonstrated at 1 to 1.5 g/bhp-hr NO_x and less than 0.05 g/bhp-hr PM. In addition, DDC has shattered the myth that natural gas will not autoignite. With their 6V-92TA "DING" engine, DDC directly injects natural gas into the engine cylinder at high pressure and autoignites the mixture without the use of spark plugs or diesel pilot injection. DDC estimates emissions from this engine to be 2 g/bhp-hr NO_x and 0.05 g/bhp-hr PM. As with alcohols, engine development for dedicated natural gas engines is lacking. Further research and development efforts will optimize these engines and determine the future for lower emission standards. Estimated costs for these engines are approximately 30 percent more than an advanced diesel with EGR and particulate trap due to increased cost of the fueling and fuel storage system. However, overall life cycle costs for natural gas is the lowest of any technology. Estimated diesel equivalent fuel economy for a lean-burn engine is about 20 percent worse than the diesel baseline. Estimated diesel equivalent fuel economy for a stoichiometric/TWC engine is approximately 30 percent worse.

Finally, electric and hybrid electric technology is just beginning to bloom. With breakthroughs in battery technology, electric buses and pick-up and delivery trucks could become zero emission vehicles by 1998. This will be of utmost importance in urban areas where pollution levels are already exceeded.

By using the above technologies, manufacturers will produce significantly cleaner heavy-duty vehicles which will play a significant role in controlling California's pervasive air pollution problems.

SECTION 2

INTRODUCTION

Oxides of nitrogen (NO_x) and particulate matter (PM) emissions from heavy-duty vehicles represent a significant portion of California's air quality problem. NO_x, along with reactive organic gases (ROG), are responsible for the pervasive smog problems in most of California's urban areas. California contains two of the nine U.S. cities, Los Angeles and San Diego, which are in extreme ozone non-attainment. As for PM emissions, a recent body of research indicates health threats from this pollutant. The South Coast Air Basin (SoCAB) exceeds the current Federal annual average PM standard by 70 percent. Heavy-duty diesel engines account for a significant portion of the controllable, inhalable, directly-emitted PM emissions from on-road motor vehicles in the SoCAB. (It should be noted that diesel engines only account for about two percent of all PM_{10} emissions in the SoCAB). If California is to attain compliance with the National Ambient Air Quality Standards (NAAQS), established by the U.S. Environmental Protection Agency (EPA) as part of the Clean Air Act (CAA), reductions in these emissions are required.

Significant improvements in diesel engine technology and fuel control have made heavy-duty engines capable of meeting today's emissions standards. The current and future Federal and California emission standards are listed in Table 2-1. In 1991, California reduced the PM emission standard for buses to 0.1 g/bhp-hr. The federal standard for buses follows in 1993. During 1991 and 1992, only alternative-fueled and trap-equipped diesel production engines have been certified to meet the 1991 California Bus Standards. However, DDC has recently certified their Series 50 engine without aftertreatment which meets this standard.

Table 2-1. California and Federal Heavy-Duty Engine Emission Standards, U.S. FTP (g/bhp-hr)

Standard	Trucks		Buses	
	NO_x	PM	NO_x	PM
1991 Federal	5.0	0.25	5.0	0.25
1991 California	5.0	0.25	5.0	0.10
1993 Federal	5.0	0.25	5.0	0.10
1994 Federal	5.0	0.10	5.0	0.07
1994 California	5.0	0.10	5.0	0.07
1996 Federal	5.0	0.10	5.0	0.05
1998 Federal	4.0	0.10	4.0	0.05

In recognizing the heavy-duty vehicle emissions problem, the California Legislature adopted Senate Bills (SB) 135 and 2330. SB 135 (California Health & Safety Code (CH&SC) 43806) requires the Air Resources Board (ARB) to adopt emissions standards and test procedures applicable to new and replacement engines for urban transit buses by January 1, 1993. These standards are to be effective by January 1, 1996, and will reflect the use of the best emission control technologies expected to be available at that time. SB 2330 (CH&SC 43701) requires the ARB to consider adopting regulations that set low emission standards for all heavy-duty vehicles. This report will be used in developing regulations for vehicles with a gross vehicle weight rating (GVWR) greater than 14,000 lb which will augment the interim transit bus regulations mandated by SB 135.

Before determining these new emission standards, ARB has contracted Acurex Environmental Corporation to provide a technical feasibility assessment of current and future heavy-duty vehicle technology. The purpose of this assessment is to identify and evaluate heavy-duty engine technologies and/or combinations of technologies that can achieve low emissions. This includes evaluation of various emission control techniques and their associated emission reductions, performance effects, and costs.

The assessment focuses on both current production and developmental engine data to demonstrate the ability of heavy-duty engines to meet substantially lower emission levels than those required by current regulations. In each section of the report, current and proposed technologies to meet low emissions goals are discussed. The report concludes with recommendations on methods for heavy-duty engines to meet low-emission goals.

2.1 HEAVY-DUTY ENGINE CLASSES

Gross vehicle weight (GVW) classifications for trucks, their GVW range, and 1990 registrations by class are shown in Figure 2-1. Both EPA and ARB divide class 2 into 2A, which has a GVW range of 6,001 to 8,500 lb, and 2B, which has a GVW range of 8,500 to 10,000 lb. Below 8,500 lb (GVW classes 1 and 2A), trucks are considered light-duty.

Three distinct weight classes of heavy-duty trucks have been established by the ARB. Light heavy-duty trucks are those with GVWs between 8,500 and 14,000 lb (truck classes 2B and 3).

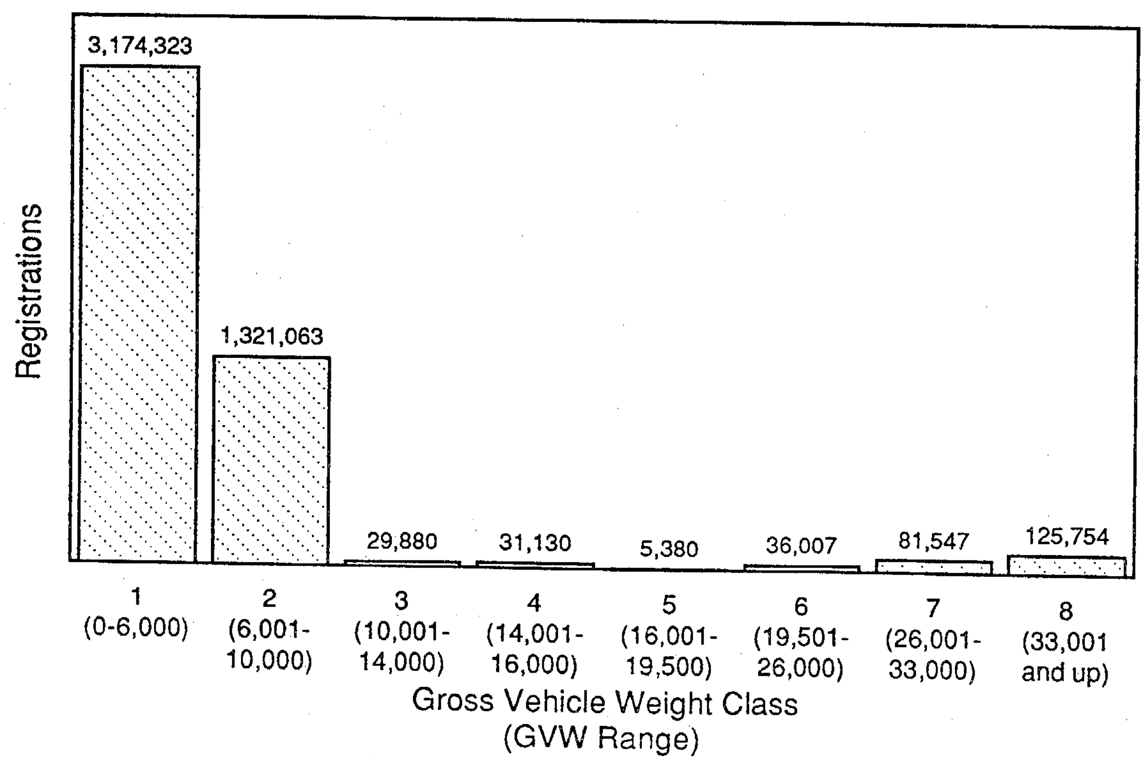

Figure 2-1. Distribution of truck registrations by GVW Class in 1990

Medium heavy-duty trucks are those with GVWs between 14,000 and 33,000 lb (truck classes 4 through 7). Heavy heavy-duty trucks are those with GVWs over 33,000 lb (truck class 8). Table 2-2 lists the heavy-duty class characteristics. However, beginning in 1995, the heavy-duty engine classification will change to GVWR of 14,000 lb or greater. In addition to the truck classes, urban transit buses are also defined by ARB. Since very stringent emissions standards have already been established for medium-duty vehicles between 6,001 and 14,000 lb GVW, transit buses are defined here as over 14,000 lb. Transit buses also must be capable of being centrally fueled. This includes conventional buses, articulated buses, express buses, shuttle buses and trailer buses. The definition excludes "over-the-road" buses which typically have separate luggage compartments, restroom facilities or long distance service. Long distance service is defined as more that 100 miles each way.

Typical vehicles that fall within the classes defined above are shown in Figure 2-2.

2.2 SIGNIFICANCE OF HDV EMISSIONS

Figures 2-3 through 2-6 show the relative contributions of ROG, carbon monoxide (CO), NO_x and PM emissions, respectively from each category of on-road motor vehicles in California for the year 2000 (Reference 1)[1]. Figure 2-7 shows the distribution of each category of motor vehicles

Table 2-2. Heavy-duty class characteristics

Characteristics	Heavy-Duty Class		
	Light	**Medium**	**Heavy**
Classes	2B-3	4-7	8
GVW Range	8,500-14,000	14,001-33,000	>33,000
General Power Range (hp)	70-170	170-250	>250
Percentage Diesel	30	56	100
Percentage Gasoline	70	44	0
California Population (1990)	85,794	154,064	125,754
Predicted 1992 VMT (mi/day)	2,862,900	13,867,000	15,735,000

[1]Numbers in parentheses refer to references found at the end of the report.

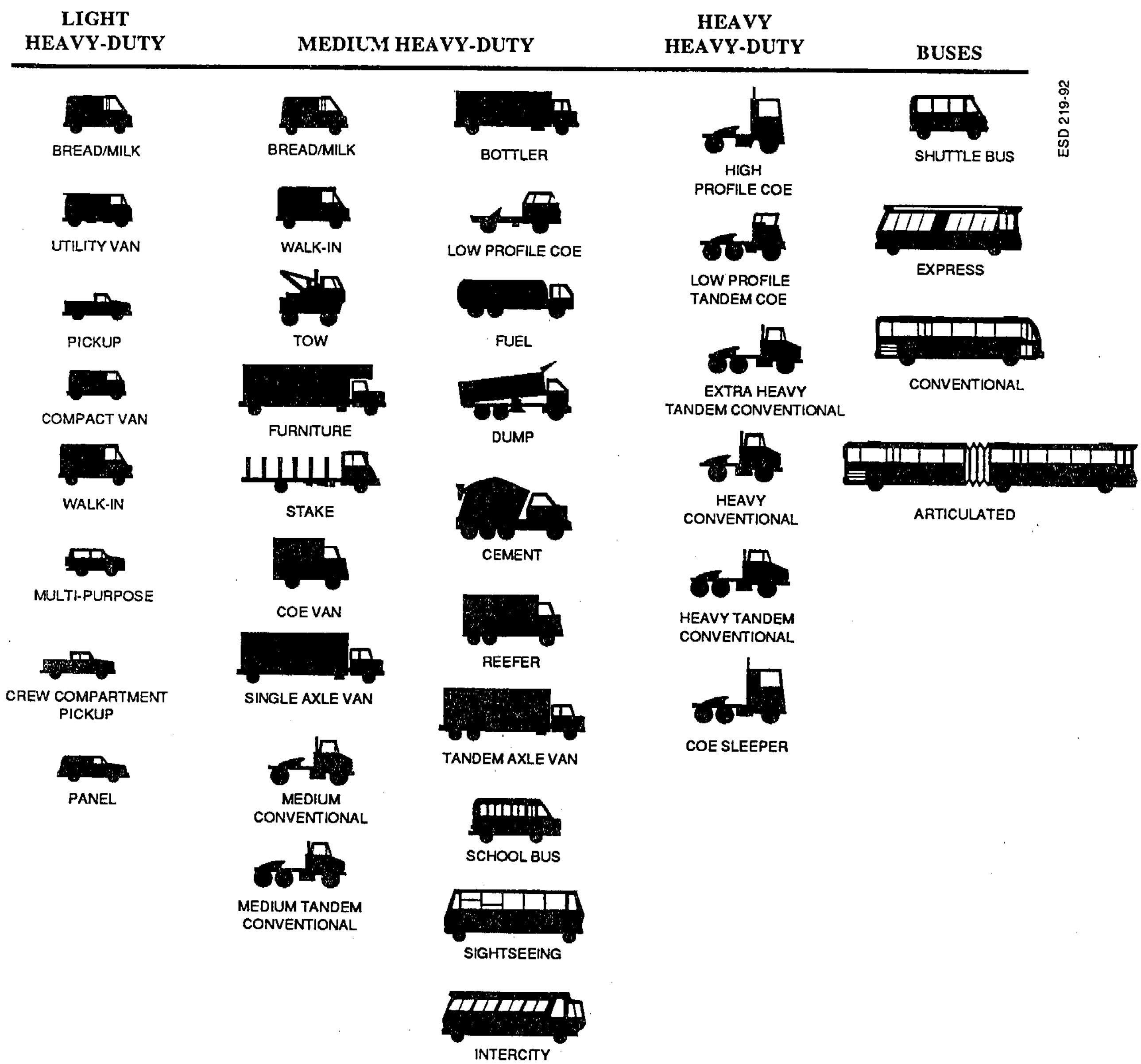

Figure 2-2. Typical heavy-duty vehicles in each heavy-duty class

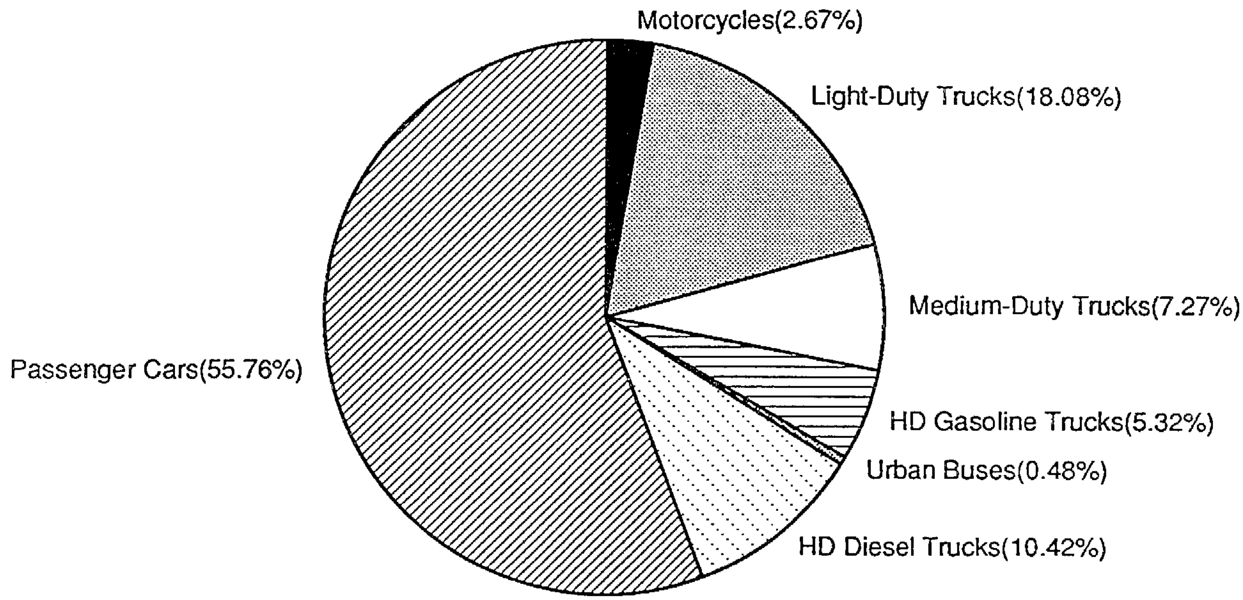

Figure 2-3. Reactive organic gas emissions distributions for on-road vehicles (2000 estimates)

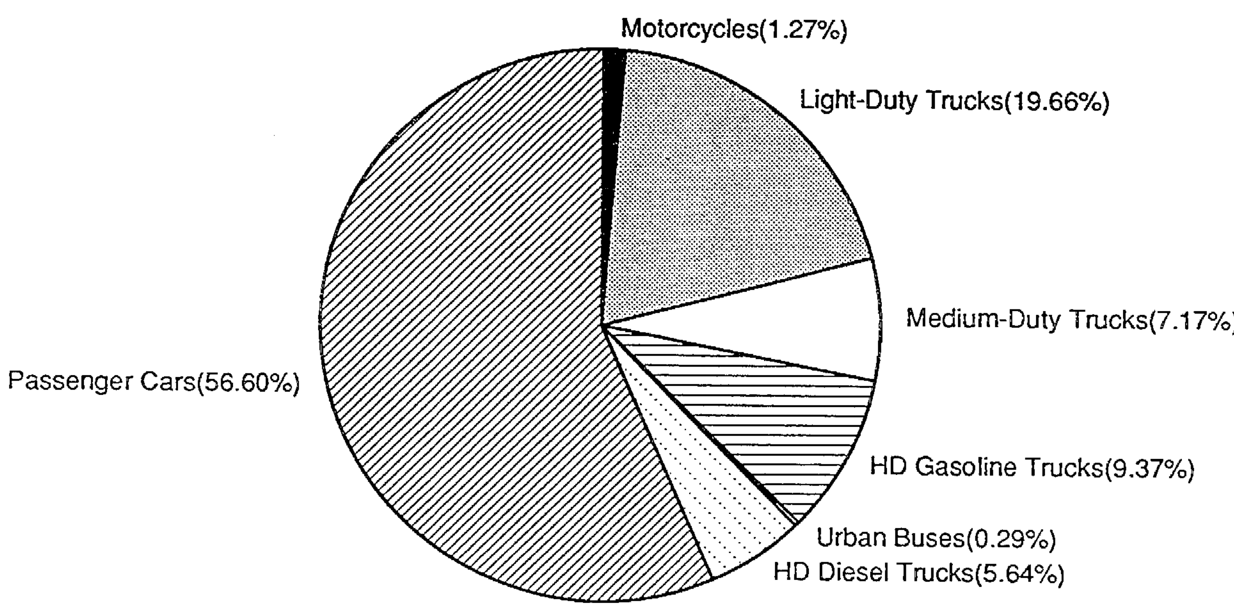

Figure 2-4. Carbon monoxide emissions distributions for on-road vehicles (2000 estimates)

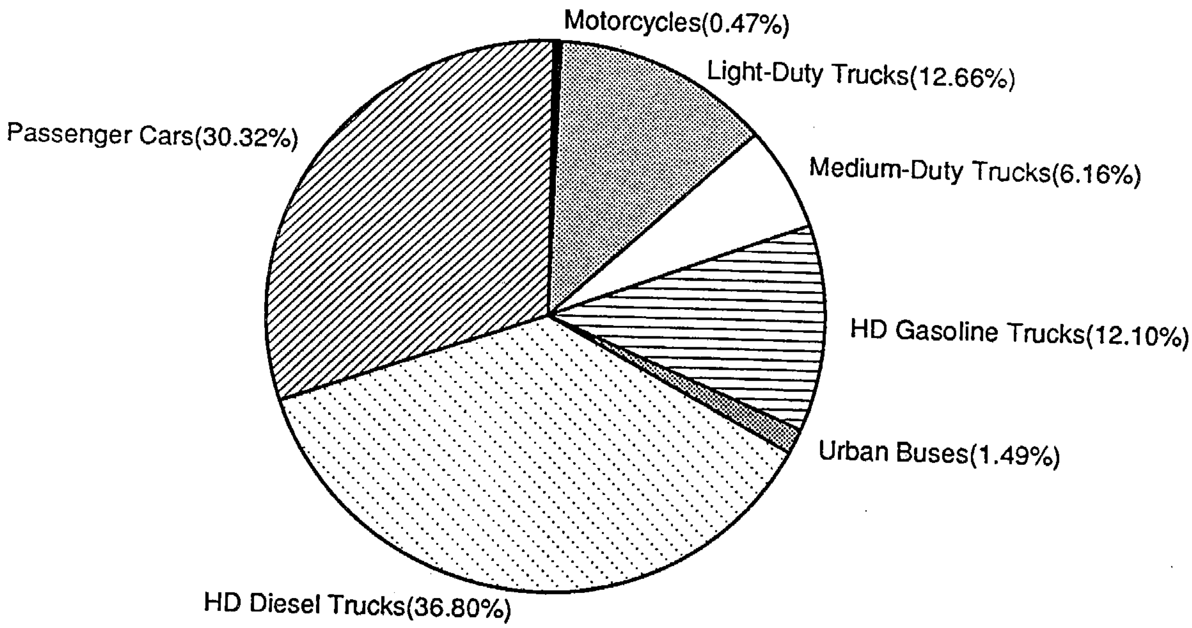

Figure 2-5. Oxides of nitrogen emissions distribution for on-road vehicles (2000 estimates)

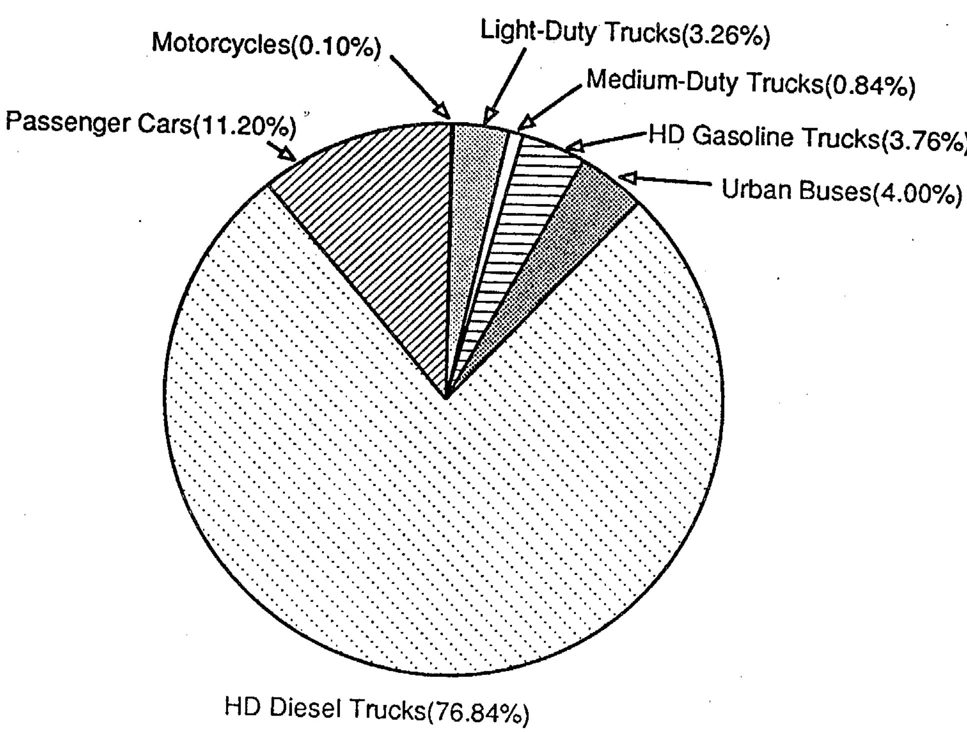

Figure 2-6. Particulate matter emissions distribution for on-road vehicles (2000 estimates)

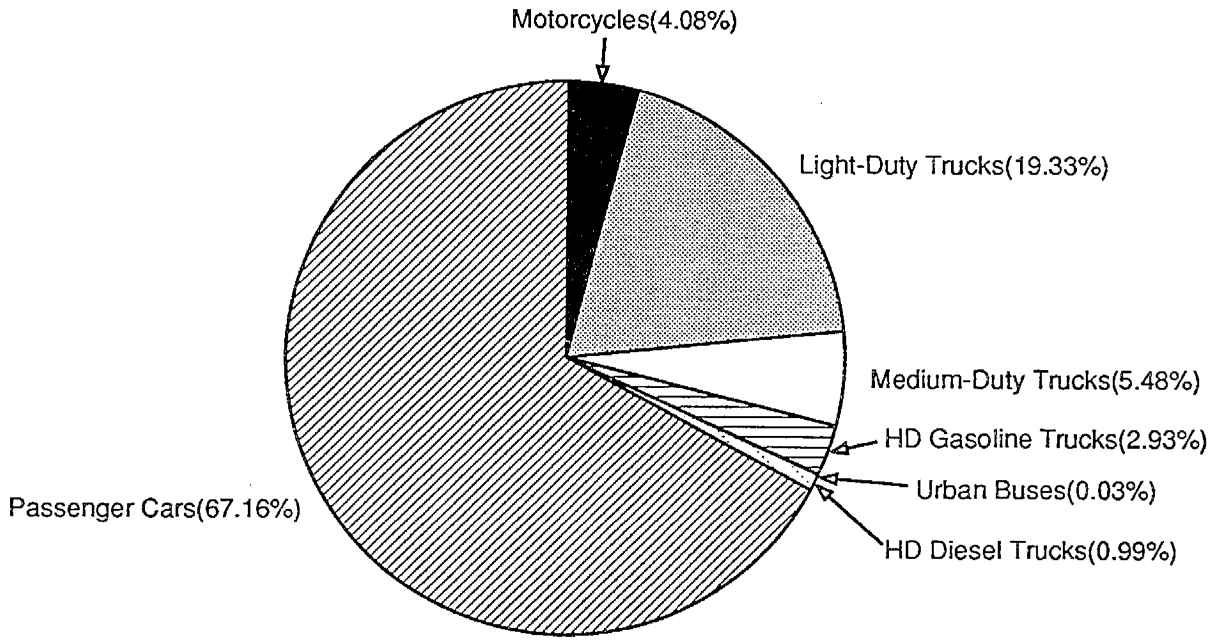

Figure 2-7. Distribution of on-road vehicles (2000 estimates)

for the year 2000. These figures show that while heavy-duty trucks and urban buses will only account for 4 percent of the vehicle population, they will contribute over 50 percent of the NO_x emissions and over 84 percent of the exhaust PM emissions of all on-road motor vehicles by the year 2000 if no further reductions in emissions standards for heavy-duty engines are enacted. The heavy-duty vehicle contribution of CO and ROG are 15.3 percent and 16.2 percent, respectively. It should be pointed out, however, that on-road mobile source emissions only account for 33 percent of the NO_x, 2 percent of the PM, 55 percent of the CO and 14 percent of the ROG inventories, respectively.

While considerable progress has been made in reducing PM emissions from heavy-duty vehicles in recent years, NO_x standards have remained relatively unchanged. The EPA will adopt a 4 g/bhp-hr standard for NO_x in 1998, but since 50 percent of the on-road mobile source NO_x emissions in California result from HDVs, significant reductions in HDV NO_x emissions must be made to clean California air.

2.3 PROPOSED EMISSION SCENARIOS

As part of SB 135 and 2330, the ARB must determine future standards for heavy-duty vehicles. SB 135 provides that regulations for urban transit buses be defined by January 1, 1993 and be effective on January 1, 1996. SB 2330 provides that regulations for all heavy-duty vehicles to be defined by December 31, 1992. The Senate bills state that the regulations should reflect the use of the best emissions control technologies expected to be available at the time the regulations become effective. The ARB is to consider the engine and fuel as a system and consider the projected cost and availability of clean burning alternative fuels and low emission vehicles compared with other air pollution control measures. Based upon current technology targets and research efforts discussed in this report, the ARB is looking to define emissions scenario goals as follows:

Scenario 1 Goals:

NO_x 2.0 g/bhp-hr

PM 0.05 g/bhp-hr

Scenario 2 Goals:

NO_x < 2.0 g/bhp-hr

PM < 0.05 g/bhp-hr

At the current time, these levels will require advances in diesel engine technology, improvements in diesel fuel quality and use of exhaust aftertreatment, or use of alternative fuels.

2.4 REPORT ORGANIZATION

Following the summary and this introduction, Section 3 describes current and future diesel engine technology. Information is provided relative to current and 1994 diesel technology, and projected future technology. Cleaner diesel fuel is also discussed.

Section 4 of this report contains an assessment of heavy-duty gasoline engine technologies including a description of reformulated gasoline. Gasoline technology is discussed in terms of present and future developments in emission control technology.

Section 5 discusses alcohol fuel technologies. Both methanol and ethanol engine technologies are discussed in detail, showing the various types of alcohol engines, their relative emissions, fuel economy, durability and future research areas.

Section 6 contains an assessment of gaseous fuel technologies, including compressed natural gas, liquefied natural gas and liquefied petroleum gas. Both current engines and research areas are discussed in this section.

Section 7 describes heavy-duty electric and hybrid electric technologies. Electric technologies have the promise of being zero emission vehicles.

Section 8 details the life-cycle costs of new technologies able to meet low emission goals.

Section 9 provides recommendations for further research areas and future demonstration programs to prove new technologies and to show durability and reliability.

Finally, Section 10 draws conclusions based upon Sections 3 through 9.

SECTION 3

DIESEL CONTROL TECHNOLOGIES

Over the last ten years considerable work has been done to reduce emissions from heavy-duty diesel engines. The diesel engine is used extensively in heavy-duty vehicle applications due to its high efficiency and superb reliability and durability. The diesel engine uses low cost fuels with reduced operating costs. However, the diesel engine also creates large amounts of oxides of nitrogen (NO_x) and particulate matter (PM) emissions in comparison to Otto-cycle engines. NO_x is formed in diesel engines from the reaction of the oxygen and nitrogen in air at high temperatures and pressures. The diesel process creates high cylinder temperatures during combustion which promotes NO_x production. PM emissions are primarily formed from incomplete combustion of heavy hydrocarbon chains as they break down during the combustion process. In addition, lubricating oil which is entrained into the exhaust or partially burned during combustion creates further PM emissions. Although hydrocarbon (HC), carbon monoxide (CO) and visible smoke emissions are also regulated, these are of less importance since control of PM emissions through more complete combustion will also control HC, CO and smoke emissions in diesel engines.

One important concept in emissions control of diesel engines is the NO_x/PM trade-off. This trade-off occurs because normal methods that reduce NO_x emissions, such as cooler peak cylinder temperatures, retarded injection timing and longer combustion durations, tend to increase PM emissions. On the other hand, shorter combustion durations tend to increase peak temperatures and pressures, thereby increasing NO_x production and limiting PM. Thus, simultaneously controlling both PM and NO_x emissions and still maintaining good fuel consumption becomes a significant challenge for diesel engine manufactures.

Control of NO_x and PM emissions within a diesel engine is linked to the rate at which the fuel is burned within the combustion chamber. Combustion rate is controlled by the rate of vaporization of the fuel droplets and the diffusion of the evaporated fuel away from fuel droplets. This is called diffusion burning. The droplet size and charge air temperature are the most important parameters in fuel vaporization rate while air utilization is the most important parameter in diffusion. By minimizing the droplet size and maximizing the fresh air to the fuel droplets, the combustion rate is maximized. Ignition delay, the time between beginning of fuel injection and the start of autoignition of the charge, is a second order effect in comparison to diffusion burning rate. Rapid burning, particularly early in the combustion process, will increase NO_x emissions. However, increased burning rate and reduced ignition delay will allow engine manufacturers to use retarded injection timing to control NO_x emissions while minimizing penalties in particulate and smoke emissions.

Advances in engine technology have made it possible for diesel engines to meet increasingly strict emission standards as shown in Figure 3-1. As emission standards become more restrictive in the coming years, other technologies such as exhaust aftertreatment, fuel reformulation and engine design improvements are being examined closely. Faced with the NO_x/PM trade-off, engine manufacturers must decide between various control options while being concerned with production costs. Such decisions will determine the future of diesel engine technology during the coming decade in California. A list of these technologies is shown in Table 3-1.

Prior to 1991, diesel engine emissions were controlled through combustion modifications and adjustments in injection timing. By retarding the injection timing, combustion occurred later in the engine cycle, reducing the peak combustion temperatures and thus reducing the amount of NO_x produced. This, however, had a negative effect on fuel economy and PM emissions. By injecting late in the cycle, peak combustion temperatures fall rapidly during the expansion stroke. This causes a quenching of combustion reactions before they are complete, thereby causing high PM emissions and high fuel consumption. It is unlikely that a 1994 technology diesel engine could be

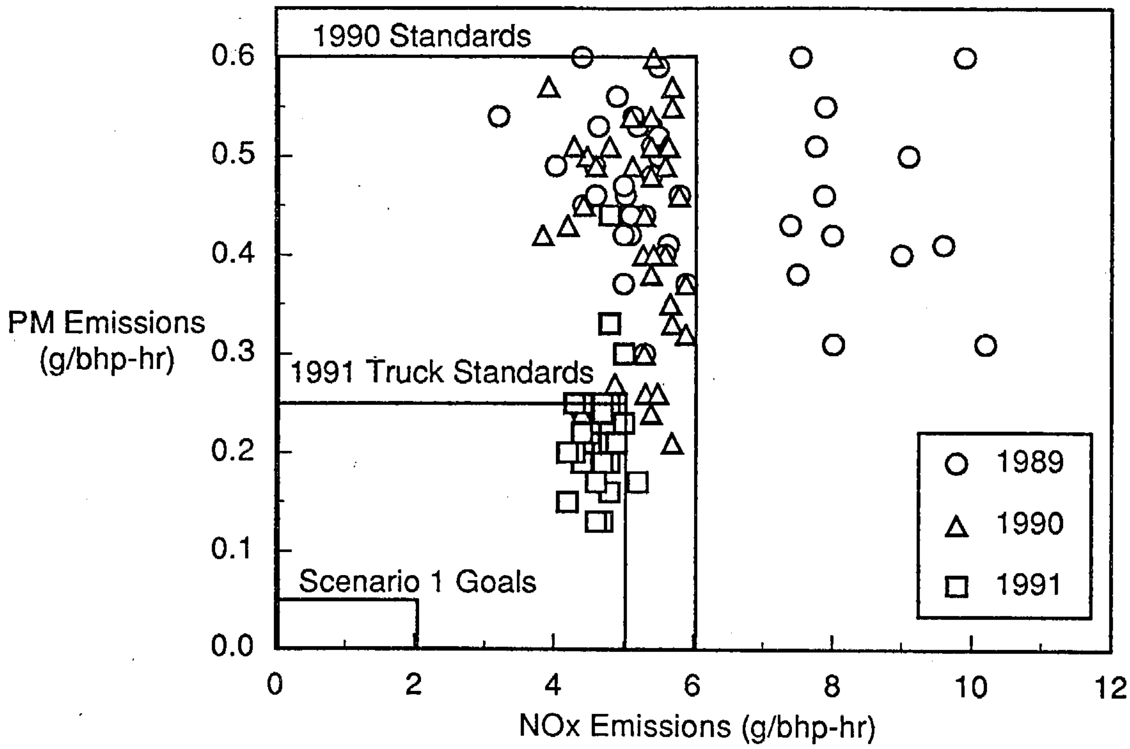

Figure 3-1. NO_x/PM emissions of current production heavy-duty diesel engines

Table 3-1. Possible diesel fueled engine technologies for current and future emission standards

1991	1994	1996+
High Pressure Fuel Injection Air-to-Air Aftercooling Fuel Injection Electronic Control Responsive Turbocharging Efficient Combustion Chambers Oil Control	Very High Pressure Fuel Injection Low Sulfur Diesel Fuel Oxidation Catalysts Particulate Traps (Buses)	Injection Rate Shaping Advanced Turbochargers Advanced Electronic Control Turbo Expander Low Inlet Manifold Temperature Exhaust Gas Recirculation Low Aromatic, High Cetane Fuel Oxygenates, Additives Advanced Traps Oxidation Catalysts DE-NO_x Catalysts

retarded enough to reduce NO_x emissions from 5 g/bhp-hr to 2 g/bhp-hr without severe misfire, unacceptable fuel consumption increases and severe penalties in cold operation. Thus additional strategies will need to be used to produce low NO_x and PM emissions simultaneously.

One strategy used to simultaneously lower NO_x and PM emissions over the federal test procedure (FTP) was electronic control of injection timing. With electronic control, injection timing could be optimized for each engine condition instead of being fixed over the entire engine range. This has a dramatic effect on NO_x and PM emissions over the FTP. It is interesting to note that several manufacturers found they were able to meet 1991 emission standards without either electronic control or exhaust aftertreatment. Many major manufacturers, however, did use electronic control to maximize fuel economy and performance while limiting emissions on several of their engine lines.

Exhaust aftertreatment has also been explored by diesel manufacturers to limit PM emissions. Several urban bus demonstration projects are currently testing particulate traps. Others are examining catalytic aftertreatment to reduce PM emissions in the exhaust. Both the DDC 6V-92TA coach engine and the Cummins L10 engine have been certified with the Donaldson dual trap system for 1992 and 1993. The use of traps is a near-term solution for some diesel-fueled engines to meet the 1991 California urban bus standards.

Diesel fuel is also being reformulated to reduce particulate emissions. In 1991, sulfur is limited to 0.05 percent by weight in the South Coast Air Basin and Ventura County. Beginning in 1993, aromatic content will be limited to 10 percent and sulfur to 0.05 percent throughout California (Smaller refiners will be allowed limit aromatics to 20 percent and other formulations which produce equivalent emissions would also be allowed). These changes in fuel quality will further reduce particulate emissions as diesel manufacturers currently certify their engines on normal low sulfur diesel certification fuels with higher aromatic content.

To better understand what can be done to lower diesel engine emissions to Scenario 1 emission levels, it is important to understand what is currently being done to reach both 1991 and

1994 emission levels. By describing efforts to reach both 1991 and 1994 emission standards, one can see that traditional methods of controlling emissions have been exhausted and that new concepts need to be developed.

In Section 3.1, the makeup of current engines that meet the 1991 emission standards is discussed. Section 3.2 contains a discussion of modifications that allow current engines to meet the 1994 standards (and the 1991 California bus standard). Section 3.3 discusses several future technologies now being researched that will allow diesel engines to approach the Scenario 1 emission goals defined in the introduction of this report. Section 3.4 summarizes design modifications and exhaust aftertreatment needed for future low emission diesel engines.

3.1 CURRENT DIESEL TECHNOLOGY

In 1991, heavy-duty engine emission standards were reduced from the 1988 levels of 6 g/bhp-hr NO_x and 0.6 g/bhp-hr PM to 5 g/bhp-hr NO_x and 0.25 g/bhp-hr PM. This required several engine design changes including fuel system, combustion chamber, and oil consumption controls. Since the rate at which NO_x is formed is a function of combustion temperature and the amount of time at which these high temperatures exist in the cylinder, reducing either combustion temperature or the residence time at high temperatures will reduce NO_x emissions. Two of the most significant methods to reduce NO_x are retarded fuel injection timing and charge air cooling. Figure 3-2 shows the effect on NO_x emissions when step changes in both injection timing and intake manifold temperature are made (Reference 2). Prior to 1991, NO_x emissions were primarily controlled by retarding injection timing and charge air cooling. Figures 3-3 and 3-4 show that retarding the injection timing increases both PM emissions and brake specific fuel consumption (BSFC). Reducing NO_x emissions below about 4.5 g/bhp-hr with injection timing retard alone will result in substantially higher BSFC and PM emissions.

Control of PM emissions in current diesel engines must focus on the composition of particulate emissions. As shown in Figure 3-5, PM in pre-1991 engines was comprised primarily of soot (or carbonaceous material), soluble fuel, soluble oil and water bound sulfates (Reference 3).

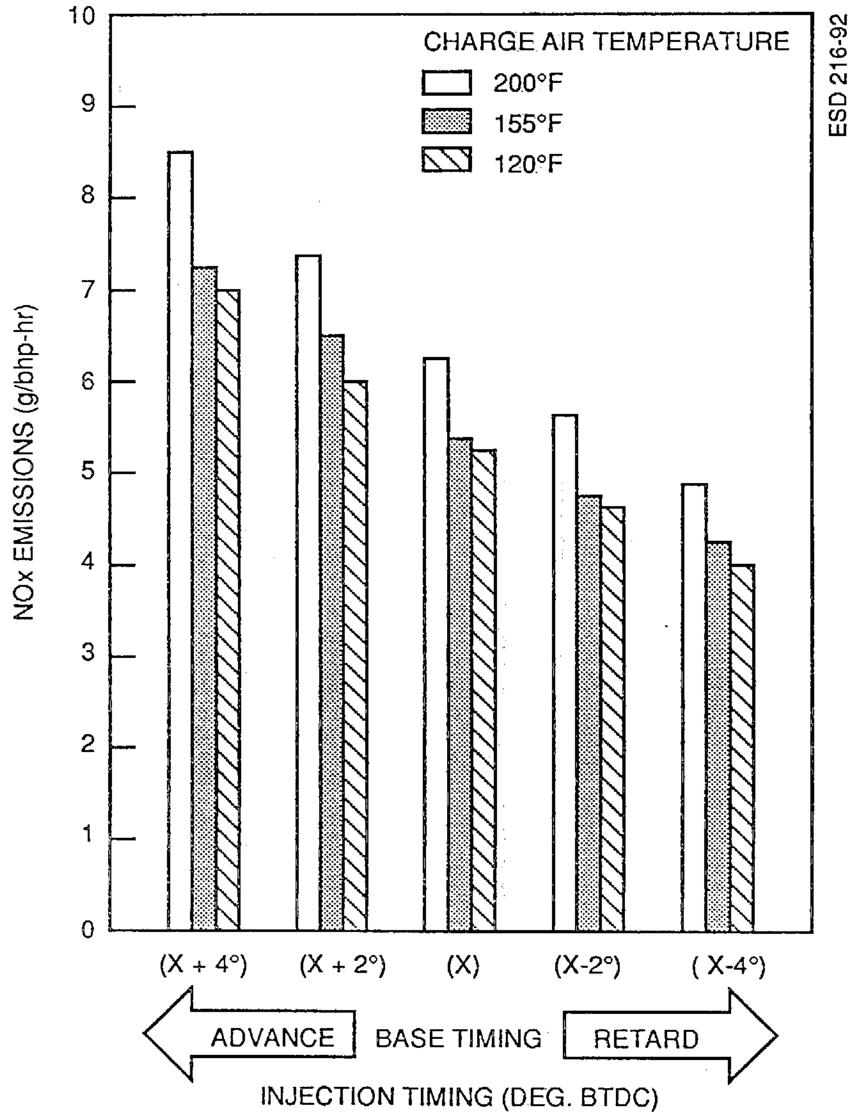

Figure 3-2. Effect of injection timing and charge air temperature on NO_x emissions

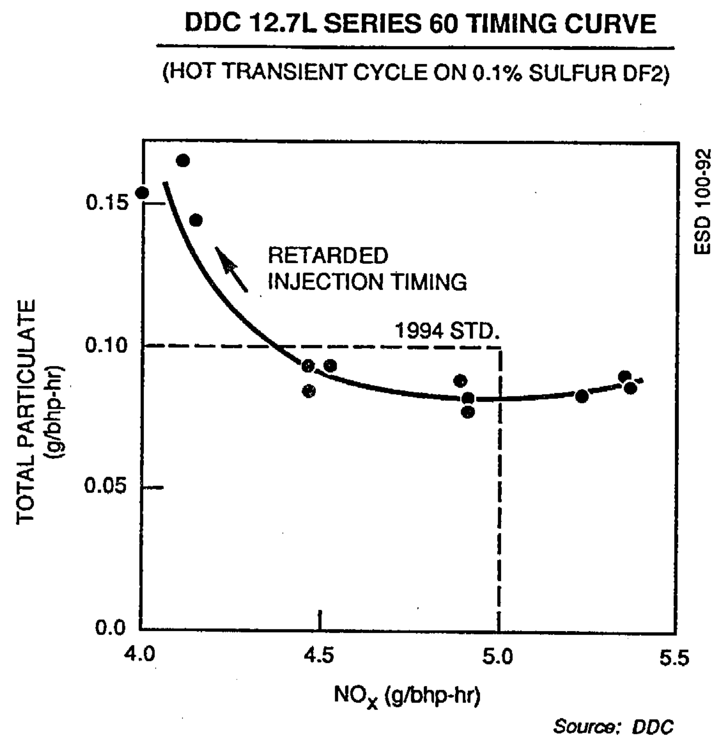

Figure 3-3. NO_x and PM emission trade-offs as a function of injection timing

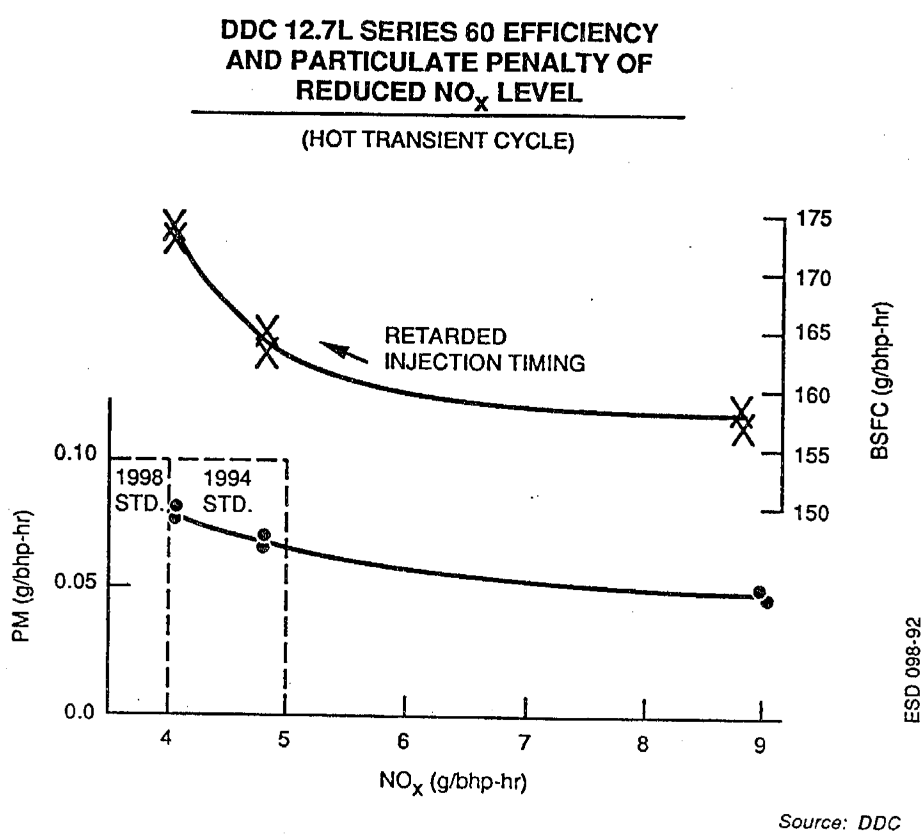

Figure 3-4. BSFC and NO_x trade-offs as a function of injection timing

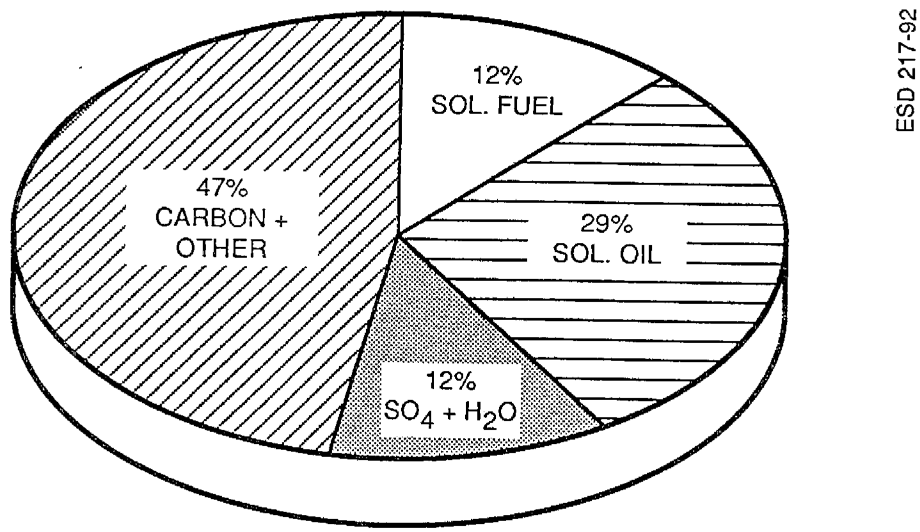

Figure 3-5. Particulate composition for a heavy-duty diesel engine with low sulfur fuel without aftertreatment

Since soot is the largest component of PM emissions, reducing soot emissions through more complete combustion significantly reduces PM emissions. This can be accomplished by carefully designing intake ports to impart organized swirl, causing the injected fuel to burn more completely within the engine cylinder. Furthermore, high pressure injection can further aid combustion by increasing vaporization and mixing, and thus increase diffusion burning rates which can improve the NO_x/PM trade-off. Reducing oil consumption while maintaining engine durability will also reduce PM emissions.

Basic progress in engine design has been the key to engines meeting the 1991 standards. These engine design changes fall into four basic categories which are discussed below. The four categories are air induction systems, combustion chamber modifications, oil control, and fuel injection systems.

3.1.1 Air Induction Systems

Pre-1991 diesel engine designs require better air-fuel management and lower intake air temperatures to meet the 1991 diesel emission standards. To accomplish this, most manufacturers added turbochargers and enhanced charge air cooling. They also modified intake manifolds and ports for better air flow and distribution. These modifications allowed most pre-1991 designs to meet 1991 emission standards without exhaust aftertreatment. The subsections below describe these modifications.

3.1.1.1 Turbochargers

Turbocharging has a major influence on the pumping losses of an engine and the combustion efficiency through control of air/fuel ratio. Increased boost at full load results in leaner mixtures, thereby significantly reducing PM emissions and increasing fuel economy. However at low speeds, air supply is critical if smoke and soot emissions are to be controlled. Using a non-optimized turbocharger at low engine speeds may not provide an adequate amount of air to control smoke. Using a non-optimized turbocharger at high speed may cause excessive boost and limit penetration of the injection plume resulting in incomplete combustion. To meet 1991 standards, some medium-duty engine manufacturers optimized the turbocharger at low speeds and used a waste-gate to reduce boost at high speeds. This resulted in reduced PM emissions and better low speed fuel economy.

3.1.1.2 Charge Air Cooling

Charge air cooling cools the intake charge and can reduce NO_x emissions. This also has a second order effect on increasing fuel economy and reducing exhaust emissions. Most 1991 medium and heavy heavy-duty engines utilize either water-to-air or air-to-air aftercooling to cool the intake charge. Air-to-air aftercooling reduces intake temperatures much more effectively than water-to-air aftercooling and reduces the thermal loading of the engine by limiting combustion temperature extremes. This increases engine life and durability. The air-to-air aftercooler can reduce NO_x emissions by up to 15 percent. The aftercooler must be designed to optimize cooling and minimize pressure drop from the turbocharger compressor. However, excessive temperature reduction at light loads can cause longer ignition delays, and thereby increase PM and HC emissions. Bypass control is used by some to limit cooling at light loads for this reason.

3.1.1.3 Intake Manifold and Port Design

Proper mixing of fuel and air in a diesel engine is essential for proper performance and emissions. Much work has been done by diesel manufacturers to optimize fuel spray penetration without allowing the spray to impinge on cylinder walls. Organized swirl deflects the fuel spray from contacting cylinder walls before combustion. Wall wetting by the fuel spray will result in high HC and PM emissions. Poor air utilization can cause rich areas to form which cannot be completely combusted. Too much deflection of the spray leads to under-penetration of the spray, also resulting in high HC and PM emission. Thus, careful attention must be used in designing intake manifolds and ports to provide an optimized amount of swirl for a given engine design and fuel injection pressure.

3.1.2 Combustion Chamber Modifications

The design of the combustion chamber is very important in defining how the fuel and air mix and how the combustion takes place. The 1991 diesel engine underwent several design changes that greatly affected emissions. One of these was the use of "reentrant" piston bowl design for some medium heavy-duty direct-injection engines (Reference 4). A diagram of a typical reentrant bowl is shown in Figure 3-6. The reentrant bowl generates vortices of swirling air eddies in the combustion bowl which assist fuel/air mixing. During expansion, additional turbulence is generated from the bowl lip which assists diffusion burning rates. Through proper design of the piston bowl, good mixing of the fuel and air can be achieved and rapid burning promoted, thereby resulting in more complete combustion and a reduction in HC and PM emissions. Many manufacturers also are reducing the top ring land area to minimize HC emissions trapped between the piston and the cylinder wall.

Higher compression ratios were also used in some 1991 diesel engines. Higher compression ratios reduce ignition delay, thereby reducing the amount of fuel burned in the premixed region and allowing more injection timing retard to control NO_x emissions. Since raising compression ratio also increases combustion temperature, cold start PM emissions and white smoke are reduced. The

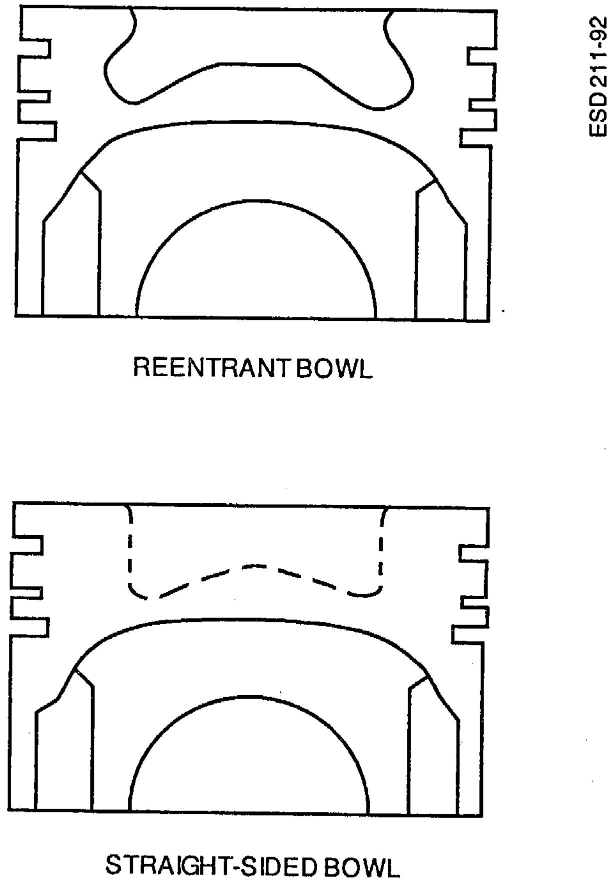

Figure 3-6. Improved combustion chamber designs

major effect, however, is at high speed, light load conditions when the ignition delay is the longest, and under cold operating conditions. In both cases, major reductions in HC emissions are achieved.

3.1.3 Oil Control

As soluble oil accounted for approximately 30 percent of pre-1991 particulate emissions from diesel engines, engine designers sought to control oil consumption without increasing engine wear. This was accomplished through improved mechanical design, cylinder bore honing and piston ring design which reduced total particulate emissions by approximately 10 percent (Reference 5).

3.1.4 Fuel Injection Systems

Perhaps the most significant improvement in diesel engines for the 1991 year was in fuel injection system control and design. Major improvements revolved around injection timing, injection pressure and injection nozzle configuration.

3.1.4.1 Injection Timing

Injection timing is the simplest and lowest cost method of controlling NO_x emissions. Unfortunately, as injection timing is retarded to reduce NO_x emissions, PM emissions and fuel consumption increase. To reduce fuel economy and PM emission penalties, engine designers have utilized higher pressure injection and higher compression ratios to reduce ignition delays and increase diffusion burning rates.

3.1.4.2 Injection Pressure

The use of high injection pressure improves the atomization and mixing of the fuel/air charge, thereby reducing the amount of soot and unburned fuel remaining at the end of combustion. This results in improved fuel economy as shown in Figure 3-7. Since increasing injection pressure decreases droplet size and increases fuel vaporization rates, diffusion burning rate is more rapid, thereby reducing PM and smoke emissions as shown in Figure 3-8. Additional injection timing retard can then be used to reduce NO_x emissions before significantly increasing smoke or fuel consumption above lower injection pressure levels.

By 1991, many manufacturers had increased injection pressure. At full-load, rated-speed conditions, typical injection pressures were increased to 15,000 to 25,000 psi from the pre-1991 levels of 10,000 to 15,000 psi.

3.1.4.3 Injection Nozzle Configuration

Diesel fuel enters the cylinder through the injector nozzle. The design and placement of the nozzle greatly affects performance and emissions of direct-injected diesel engines. Significant research has been done to optimize the diesel nozzle spray cone and inclination for the 1991 diesel engines. As a result the following conclusions were reached:

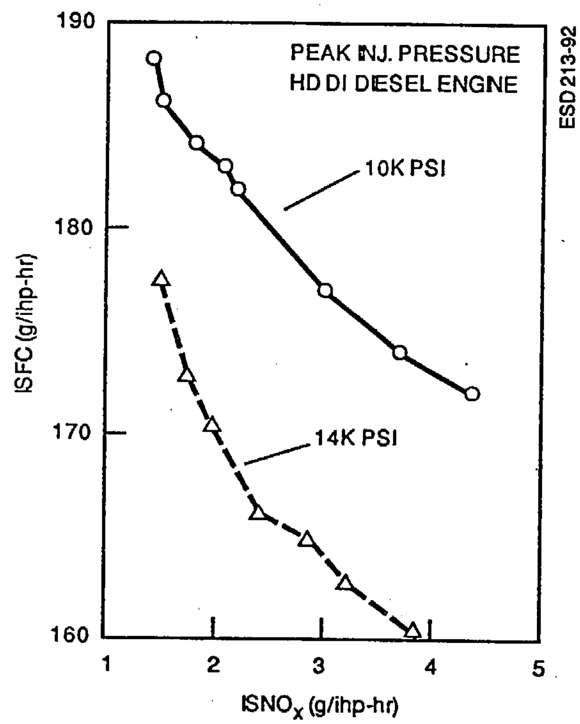

Figure 3-7. Effect of injection pressure on NO_x and fuel consumption

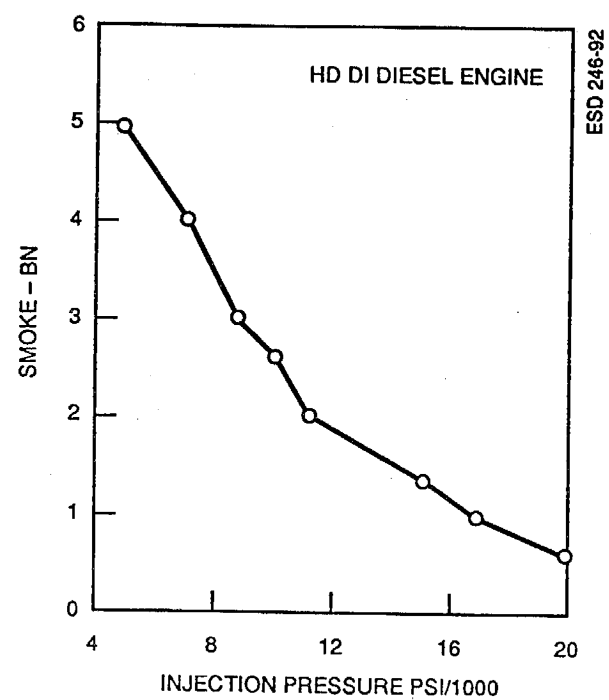

Figure 3-8. Effect of injection pressure on smoke emissions

- Nozzle sac volumes should be minimized to reduce HC emissions. Fuel remaining in the dead volume at the end of the injector (sac volume) can dribble out after injection, resulting in high gaseous hydrocarbon emissions. Minimization of this volume reduces hydrocarbon emissions, but this effect tends to be lost in larger engines due to the brake specific emission effects (g/bhp-hr) as shown in Figure 3-9. Valve-covered-orifice (VCO) nozzles further minimize fuel leakage by covering injection nozzle holes when not injecting fuel, but tend to greatly increase stresses on valve seats. VCO nozzles are not strictly required for heavy-duty 1991/4 engines due to the brake specific emission effects.
- The injector nozzle holes should be optimized for the specific combustion chamber as well as for the injection pressure to maximize mixing with the cylinder air. The number of holes, hole diameter and length of the nozzle hole must be properly selected to provide proper fuel atomization and spray penetration.
- The spray cone and inclination should also be optimized for the specific combustion system. Again, proper spray cone definition and inclination can maximize air utilization (thereby minimizing PM and HC emissions) and reduce wall wetting. Since many medium-duty and some heavy-duty engines currently use two valves per cylinder and an offset and inclined injector mounting, significant improvement in combustion efficiency can be realized by utilizing a more central and vertical injection nozzle.

Electronic control allows adjustable injection timing which can be changed rapidly for transient load, speed and temperature requirements. Better control of injection duration can limit particulate and smoke emissions. Current and near-term electronic fuel injection systems capable of meeting low emission goals are listed below.

- Advanced rotary pump with electronic control systems are being developed for smaller medium-duty engines. Their limitations are the amount of fuel which can be injected

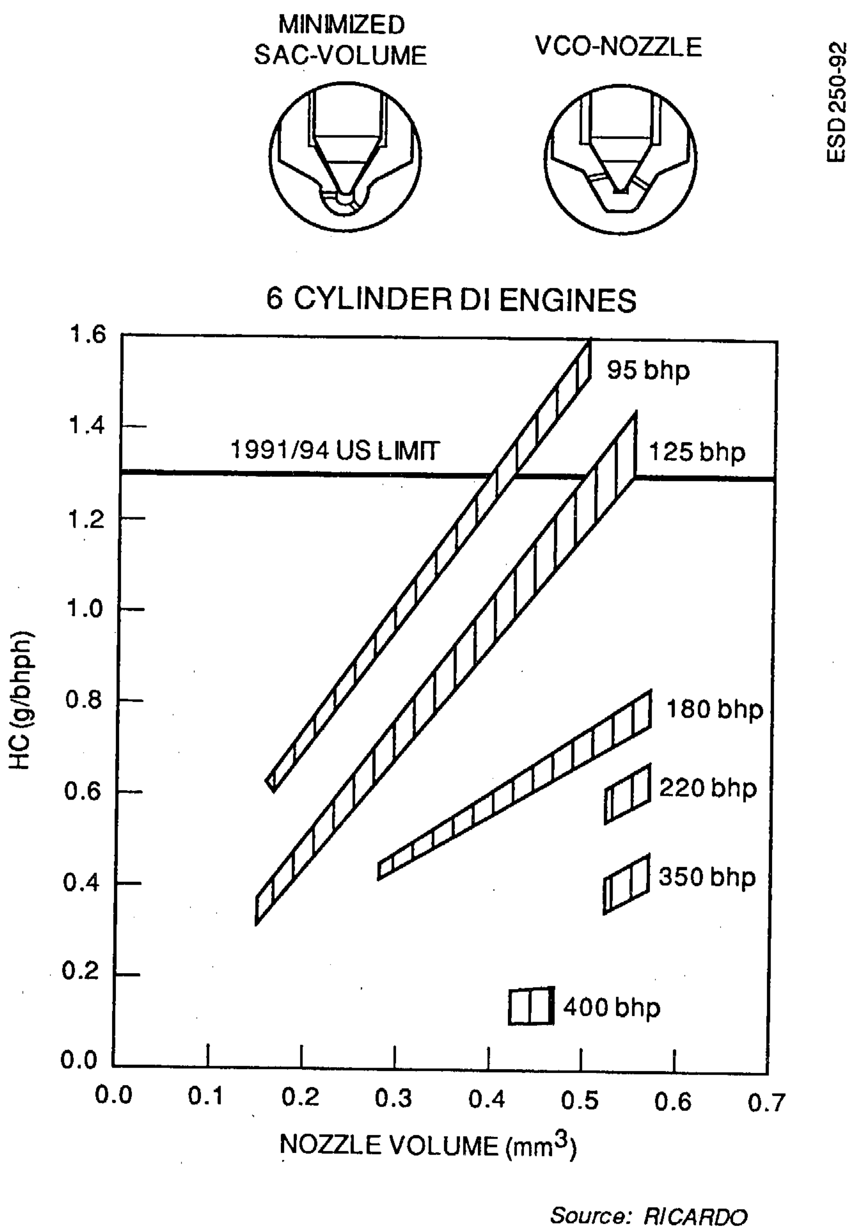

Figure 3-9. Effect of injector sac volume on hydrocarbon emission

(thus limiting its use to less than 30 bhp/cylinder engines) and maximum injection pressure (less than 16,000 psi). Rapid spill systems are being developed.

- Electronic sleeve control in-line fuel pumps are suited for medium and heavy-duty engines. It is possible to reach 22,000 psi injection pressure with this system. This system can give excellent low speed injection pressures and rapid spill.
- Electronic unit injectors (EUI) are becoming the most dominant system for heavy-duty engines and some medium-duty engines. EUI are mechanically driven with electronic control. EUI control both the beginning and end of injection by use of a solenoid valve as shown in Figure 3-10. However, EUI were not strictly necessary for 1991, but were adopted by many manufacturers for other reasons, such as cost and future potential. A mechanical fixed prelift in-line pump is adequate for 1991.

3.1.5 Current Technology Summary

As shown in Tables 3-2, 3-3 and 3-4, most of the light, medium and heavy heavy-duty engines are turbocharged with air-to-air aftercooling. Most of the light and medium heavy-duty engines still maintain mechanical fuel injection, while most of the heavy heavy-duty engines have moved to electronics. All of the engines listed required either engine modifications or electronic fuel injection systems to reach 1991 emission levels, while some required both. Figure 3-11 shows these engines on a NO_x versus PM plot.

Some of the engines shown in Figure 3-11 exceed the PM standard. EPA allows some engines to exceed the standard if the amount exceeded is offset by heavy-duty truck engines that volunteer to meet a lower standard. ARB, however, only allows PM averaging of alternative-fueled urban buses certified to meet 0.1 g/bhp-hr PM with petroleum-fueled non-urban bus engines through the 1995 model year.

To reach 1991 emission levels, engine manufacturers have utilized most of the traditional NO_x control techniques. Further injection timing retard would severely impact BSFC and PM emissions. Further improvements in NO_x and PM emissions will require some further advances in

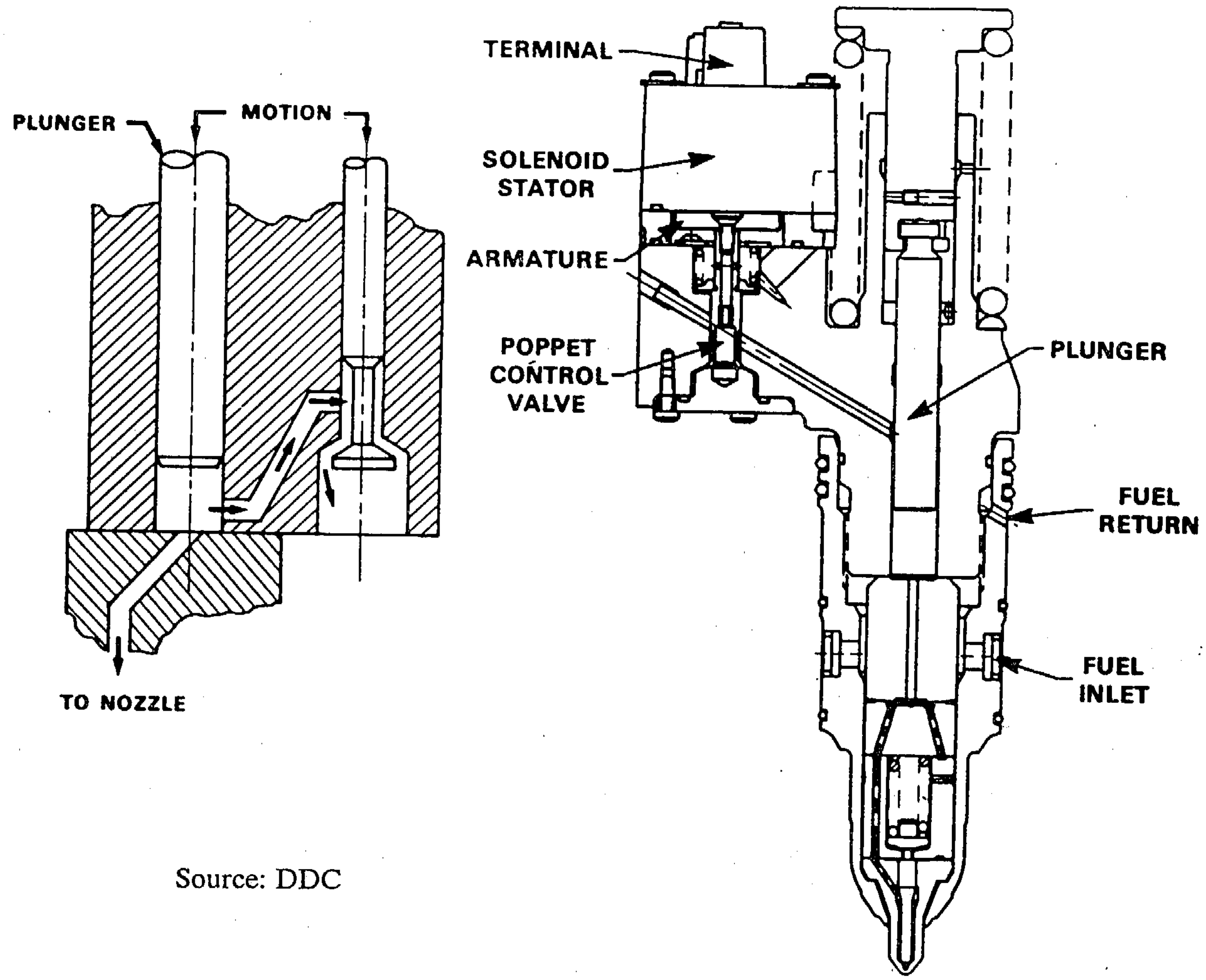

Figure 3-10. Diagram of an electronic unit injector

Table 3-2. 1992 EPA certified data for light heavy-duty diesel engines[1]

Manufacturer	Engine Model	Cyl	Disp (l)	Fuel System	Aspiration[2]	Service Class	Emission Control[3]	HP @ Speed	Torque @ Speed	Emissions (g/bhp-hr)			
										HC	CO	NO_x	PM
General Motors	V-8 6.2L,IDI	V-8	6.2	Mechanical	Natural	Light	IDI/EM	165@3500	275@2160	0.4	2.2	4.1	0.20
General Motors	V-8 6.5L,IDI	V-8	6.5	Mechanical	TC	Light	IDI/EM	200@3400	385@1700	0.4	2.0	3.7	0.16
Hino	W04C-TH	I-4	3.8	Mechanical	TCA, A/A	Light	EM	150@2800	304@1800	0.4	1.3	4.2	0.15
Isuzu	4BD2TC	I-4	3.9	Mechanical	TCA, A/A	Light	EM	129@3000	245@1900	0.3	0.8	4.0	0.16
Mitsubishi	4D34-1AT3	I-4	3.9	Electronic	TCA, A/A	Light	EM/EC	135@3000	253@1800	0.3	2.4	4.3	0.20
Mitsubishi	6D31-1AT2	I-6	5.0	Electronic	TCA, A/A	Light	EM/EC	155@3000	289@2000	0.5	1.9	4.6	0.16
Navistar	7.3/A185	V-8	7.3	Mechanical	Natural	Light	EM	185@3300	345@1980	0.6	--[4]	4.4	0.24
Nissan	TD42T	I-6	4.2	Mechanical	TC	Light	EM	135@3500	217@2000	0.2	0.6	4.2	0.19
Perkins	Phaser 110TI	I-4	4.0	Mechanical	TCA, A/A	Light	EM	110@2600	281@1600	0.4	1.3	4.8	0.19

[1]Source: EPA Federal Certification Test Tesults for 1992 Model Year.
[2]TC = turbocharged.
TCA = turbocharged and aftercooled.
A/A = air to air intercooler.
W/A = water to air intercooler.
[3]EM = engine modifications.
EC = electronic control.
[4]CO waiver.

Table 3-3. 1992 EPA certified data for medium heavy-duty diesel engines[1]

Manufacturer	Engine Model	Cyl	Disp (l)	Fuel System	Aspiration[2]	Service Class	Emission Control[3]	HP @ Speed	Torque @ Speed	Emissions (g/bhp-hr)			
										HC	CO	NO_x	PM
Caterpillar	3116	I-6	6.6	Mechanical	TCA, A/A	Medium	EM	185@2600	520@1560	0.6	0.8	4.6	0.15
Caterpillar	3116	I-6	6.6	Mechanical	TCA, A/A	Medium	EM	290@2600	732@1560	0.2	1.3	4.4	0.19
Cummins	B3.9-120	I-4	3.9	Mechanical	TCA, A/A	Medium	EM	120@2500	300@1700	0.3	1.6	4.3	0.33
Cummins	B5.9-160	I-6	5.9	Mechanical	TCA,A/A	Medium	EM	160@2500	400@1700	0.4	1.2	4.9	0.24
Cummins	B5.9-230	I-6	5.9	Mechanical	TCA, A/A	Medium	EM	230@2500	605@1600	0.2	1.1	4.1	0.16
Ford	270	I-6	7.8	Mechanical	TCA, A/A	Medium	EM	270@2300	740@1500	0.3	1.9	4.6	0.22
Hino	H06C-TP	I-6	6.5	Mechanical	TCA, A/A	Medium	EM	180@2500	448@1500	0.8	1.3	4.3	0.17
Isuzu	6BG1XN	I-6	6.5	Electronic	TCA, A/A	Medium	EM/EC	172@2600	376@1500	0.7	2.1	4.3	0.19
Mercedes-Benz	OM 366 LA	I-6	5.9	Mechanical	TCA, A/A	Medium	EM	230@2600	590@1400	0.4	1.4	4.8	0.20
Mercedes-Benz	OM 447 LA	I-6	12.0	Mechanical	TCA, A/A	Medium	EM	410@2100	1327@1100	0.4	1.1	4.7	0.24
Mitsubishi	6D16-1AT2	I-6	7.5	Mechanical	TCA, A/A	Medium	EM/EC	220@2600	470@1600	0.3	1.1	4.5	0.17
Navistar	DTA-360/ C190TF	I-6	5.9	Mechanical	TCA, A/A	Medium	EM	190@2700	485@1700	0.7	2.6	4.7	0.29
Navistar	DTA-466/ E70F	I-6	7.6	Mechanical	TCA, A/A	Medium	EM	270@2400	800@1600	0.4	1.9	4.4	0.32
Nissan	FE6TA	I-6	6.9	Mechanical	TCA, A/A	Medium	EM	210@2800	434@1800	0.8	1.9	4.0	0.20
Perkins	Phaser 180TA	I-6	6.0	Mechanical	TCA, A/A	Medium	EM	180@2600	450@1600	0.4	1.5	4.8	0.21
Volvo White	TD73EA	I-6	6.7	Mechanical	TCA, A/A	Medium	EM	215@2400	610@1400	0.4	1.7	4.6	0.13

[1]Source: EPA Federal Certification Test Tesults for 1992 Model Year.
[2]TC = turbocharged.
TCA = turbocharged and aftercooled.
A/A = air to air intercooler.
W/A = water to air intercooler.
[3]EM = engine modifications.
EC = electronic control.

Table 3-4. 1992 EPA certified data for heavy heavy-duty diesel engines[1]

Manufacturer	Engine Model	Cyl	Disp (l)	Fuel System	Aspiration[2]	Service Class	Emission Control[3]	HP @ Speed	Torque @ Speed	Emissions (g/bhp-hr) HC	CO	NO_x	PM
Caterpillar	3176	I-6	10.3	Electronic	TCA, A/A	Heavy	EM	325@1800	1250@1200	0.2	3.1	4.5	0.20
Caterpillar	3306C	I-6	10.5	Mechanical	TCA, A/A	Heavy	EM	300@2000	1100@1200	0.6	1.3	4.2	0.24
Caterpillar	3406B PEEC	I-6	14.6	Mechanical	TCA, A/A	Heavy	EM	460@1900	1650@1200	0.3	1.7	4.7	0.21
Cummins	L10-310E	I-6	10.0	Electronic	TCA, A/A	Heavy	EC	310@1600	1150@1200	0.3	3.1	3.9	0.28
Cummins	L10-280E	I-6	10.0	Electronic	TCA, A/A	Heavy	EC	280@2000	900@1200	0.3	2.9	4.3	0.23
Cummins	N14-460E	I-6	14.0	Electronic	TCA, A/A	Heavy	EC	460@1700	1550@1220	0.3	2.3	4.5	0.15
Cummins	N14-370	I-6	14.0	Mechanical	TCA, A/A	Heavy	EM	370@1600	1400@1200	0.6	1.9	4.1	0.20
Detroit Diesel	6L-71TA DDEC	I-6	7.0	Electronic	TCA, W/A	Heavy	EM/EC	330@2100	1050@1200	0.3	1.9	4.9	0.20
Detroit Diesel	6L-71TA DDEC Coach	I-6	7.0	Electronic	TCA, W/A	Heavy	EM/EC	270@2100	880@1200	0.3	1.9	4.7	0.22
Detroit Diesel	6V-92TA DDEC	V-6	9.0	Electronic	TCA, W/A	Heavy	EM/EC	350@2100	1020@1200	0.4	2.0	4.8	0.21
Detroit Diesel	6V-92TA DDEC Coach	V-6	9.0	Electronic	TCA, W/A	Heavy	EM/EC	277@2100	880@1200	0.4	1.8	4.8	0.22
Detroit Diesel	6V-92TA DDEC Coach	V-6	9.0	Electronic	TCA, W/A	Heavy	EM/EC	253@2100	775@1200	0.4	2.1	4.8	0.25
Detroit Diesel	Series 60 11.1L	I-6	11.1	Electronic	TCA, A/A	Heavy	EM/EC	350@1800	1250@1200	0.1	2.2	4.7	0.13
Detroit Diesel	8V-92TA DDEC	V-8	12.1	Electronic	TCA, W/A	Heavy	EM/EC	500@2100	1470@1200	0.3	2.7	4.6	0.29
Detroit Diesel	Series 60 12.7L	I-6	12.7	Electronic	TCA, A/A	Heavy	EM/EC	450@2100	1450@1200	0.1	1.7	4.8	0.17
Mack	EM7-300	I-6	11.9	Electronic	TCA, A/A	Heavy	EM/EC	300@1750	1425@1020	0.1	--[4]	4.4	0.24
Mack	E7-400	I-6	11.9	Electronic	TCA, A/A	Heavy	EM/EC	400@1800	1460@1250	0.1	--[4]	4.8	0.15
Saab-Scania	DSC11-38	I-6	11.0	Mechanical	TCA, A/A	Heavy	EC	336@1900	1130@1000	0.4	1.1	4.6	0.18
Volvo White	TD123EB	I-6	12.0	Electronic	TCA, A/A	Heavy	EM/EC	330@1900	1250@1200	0.4	1.6	4.8	0.21

[1]Source: EPA Federal Certification Test Tesults for 1992 Model Year.
[2]TC = turbocharged.
TCA = turbocharged and aftercooled.
A/A = air to air intercooler.
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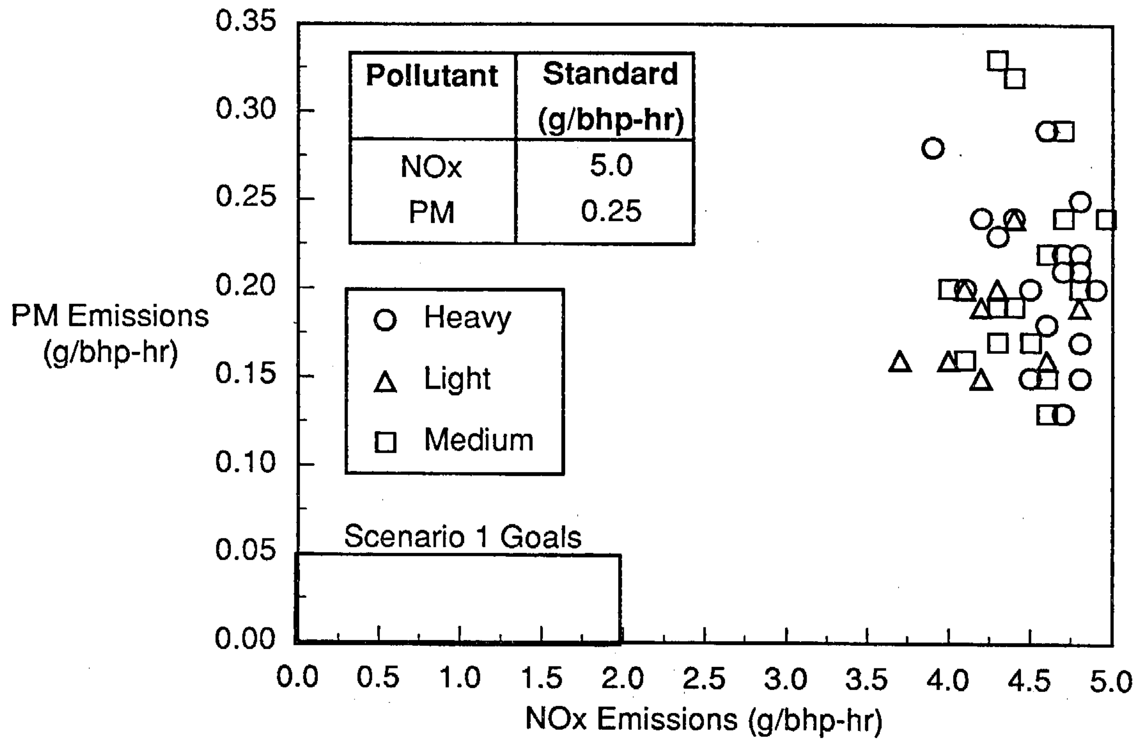


Figure 3-11. NO_x and PM emissions for EPA certified 1992 diesel engines

technology, such as combustion chamber modifications, better air utilization, use of exhaust aftertreatment, and diesel fuel reformulation.

3.2 1994 DIESEL TECHNOLOGY

As 1994 approaches, diesel manufacturers are seeking low-cost methods of meeting the even stricter particulate standard of 0.1 g/bhp-hr. Since many of the more fundamental engine design modifications were used to meet the 1991 standards, manufacturers are examining exhaust aftertreatment to further reduce particulate emissions. However, very high pressure fuel injection and further combustion chamber, turbocharger and aftercooler optimization are also being investigated to prevent the need of particulate traps for most engines. Some bus engines will most likely require particulate traps or catalytic converters to meet the 1994 emission standards (or 1991 California urban bus standards).

In this section, further improvements to meet the 1994 emission standards will be described. These improvements include very high pressure fuel injection, air management, combustion chamber design, cleaner diesel fuels and exhaust aftertreatment. The two kinds of exhaust aftertreatment discussed in this section are particulate traps and oxidation catalysts.

3.2.1 Further Fuel Injection Improvements

Increasing fuel injection pressure is one of the most effective means of reducing particulate emissions. Higher injection pressure results in better atomization, and therefore, better air utilization and reduced ignition delay. By increasing the fuel injection pressure from 15,000 psi to 20,000 psi, the PM versus NO_x injection timing curve can be lowered. At the same NO_x level of 5 g/bhp-hr, PM is reduced from 0.25 g/bhp-hr to 0.15 g/bhp-hr for this change in injection pressure. While fuel injection components need to be strengthened to handle the increased pressure, resulting in increased cost, many manufacturers have found very high pressure injection to be a cost-effective means of reducing PM emissions. With this approach, a quiescent or very low swirl combustion chamber can be used allowing the mixing process to be controlled by injection pressure.

Another approach is to use moderate injection pressures and higher swirl to improve air utilization and diffusion burning rate. The swirl assisted system uses a reentrant piston bowl to enhance swirl and squish and thereby improve mixing. This is much more cost effective for lower cost medium-duty engines.

3.2.2 Further Air Management Improvements

Many manufacturers found that more precise turbocharger matching to engine conditions provided significant decreases in part load particulate emissions. In addition, many manufacturers converted from water-to-air aftercooling to air-to-air aftercooling, thereby reducing air inlet temperatures approximately 100°F and significantly decreasing NO_x emissions. This allowed manufacturers to set injection timing for lower particulate emissions without exceeding 5.0 g/bhp-hr NO_x.

3.2.3 Further Combustion Chamber Improvements

With increased injection pressure, many manufacturers found that intake port swirl had to be reduced. Too much swirl can cause excessive mixing which causes the charge to burn more vigorously and increase NO_x formation. It is important to match swirl with injection pressure to minimize NO_x and particulate emissions in direct-injected diesel engines.

3.2.4 Cleaner Diesel Fuel

Fuel quality and composition can greatly affect particulate emissions. Fuel properties most often shown to reduce PM emissions are cetane number, aromatic content and sulfur content.

Increasing cetane number can reduce ignition delay, thereby reducing pressure rise rates and peak temperatures, which in turn produces less NO_x emissions. NO_x emissions have been shown to decrease by as much as 10 percent when cetane number was increased from 40 to 53. Particulate, HC, and CO emissions also decrease due to more complete combustion, especially at low speeds, and there is a greater avoidance of lean flame-out conditions. For the cetane number increase from 40 to 53, PM emissions decreased 10 percent, CO emissions decrease 56 percent, and HC emission decrease 43 percent (Reference 6).

Aromatics typically comprise from 25 to 45 percent of No. 2 diesel fuel by volume. These aromatics contribute disproportionately to formation of soot emissions. High aromatic diesel fuels do not burn as readily and are more inclined to pyrolyze into particulate emissions. Reducing aromatic content from 40 to 10 percent can reduce particulate emissions approximately 10 percent (Reference 7).

Sulfur in diesel fuel results in sulfate particulates that bind with water vapor and are counted as part of the particulate emissions from diesel engines. Sulfur dioxide emissions from diesel engines form sulfate particulates in the atmosphere and are considered secondary particulate emissions. Sulfur also affects the performance of some catalytic diesel particulate traps and diesel oxidation catalysts. Reducing sulfur content from 0.29 to 0.05 percent reduces particulate emissions 19 percent (Reference 9).

California has legislated lower aromatic content and sulfur content for diesel fuel. Effective October 1993, all diesel fuel sold in California will contain only 0.05 percent sulfur and a lower aromatic content (10 percent for large refineries and 20 percent for small refineries). However additional reductions in sulfur content may be required to further reduce particulate emissions, especially if catalytic converters need to be used.

3.2.5 Exhaust Aftertreatment

Particulate composition can be divided into insoluble particulate and soluble particulate. Insoluble particulates are mainly carbonaceous, sulfate and ash particulates. Soluble particulates are partially burned lube oil and fuel-based particulates or aerosols.

Particulate traps capture and periodically incinerate (known as regeneration) carbonaceous exhaust particulates. Additional heat must be added to accomplish regeneration since exhaust temperatures are not high enough to perform this task consistently in most duty cycles. Most commonly, electric heaters or burners are used, but a variety of other methods have been utilized. Regeneration is triggered after reaching a pre-established pressure drop measured across the trap. More sophisticated systems also monitor trap core temperature and air flow rate to determine the need to regenerate.

Oxidation catalysts are another method of reducing exhaust particulates. These are similar to catalysts used on gasoline engines, however, diesel catalysts must operate at lower temperatures and oxidize heavier hydrocarbon species. Oxidation catalysts are only feasible with low sulfur fuels. Using high sulfur fuels with oxidation catalysts leads to the formation of sulfates which increase particulate emissions rather than decrease them.

A comparison of traps and oxidation catalysts is shown in Table 3-5.

3.2.5.1 Particulate Traps

Traps are the most effective method of reducing particulate emissions, with efficiencies between 60 and 90 percent. There are a large variety of materials and approaches for trapping particulate (which is relatively easy) and regenerating the trap. The latter represents the most

Table 3-5. Comparison between particulate traps and oxidation catalysts for diesel particulate control

Particulate Traps	Oxidation Catalysts
Burns carbonaceous particulates	Reduces SOF
Auxiliary regeneration required—expensive controls	No auxiliary regeneration required—no controls
Durability/reliability concerns	Deactivation concerns
Cost reduction opportunities	Cost effective
Sensitive to temperature gradient	Sensitive to sulfur in fuel

difficult aspect of particulate trap development. Regeneration adds significant cost and complexity to the particulate trap system. The most representative trapping methods can be split into two broad groups as follows:

- Trap with on-board regeneration (e.g., burner) system. Systems have been certified for city bus applications. On-board regeneration is required for the uncertain duty cycles associated with on-road operation. The major difficulties for the future are high cost and unproven reliability and durability of these systems. Alternatives to the use of a burner are electric heating, increasing engine temperature for catalytic regeneration, and additives which can be sprayed on the trap or added to fuel. Some concerns exist over the environmental effects of these additives which contain metals such as copper.
- Trap regenerated at central depot. An example of this approach is the Volvo Cityfilter which features electrical heating and regeneration overnight when the vehicle is out of use. This approach is suitable for light-duty and some medium-duty applications and is a relatively low cost approach.

Several types of traps with on-board regeneration are being demonstrated in the U.S. Different trap technologies are summarized in Table 3-6. Various regeneration approaches and filter types are indicated in this table. Diesel soot requires temperatures around 1,100°F to

Table 3-6. Particulate trap technologies

Manufacturer	Filter Element Type	Regeneration Type	Number of Filter Elements
Donaldson Dual Electric Regeneration	Ceramic monolith	Electrically heated filter of bypassed trap	2
ORTECH/Donaldson Dual Burner Regeneration	Ceramic monolith	Diesel burner heats air over bypassed trap	2
3M fiber coil trap	Nextel woven ceramic fibers	Electric heating of alternating elements	12 to 16
Asahi Ceramic Wall Flow	Ceramic monolith	Pulse cleaning of filter with electric burning of collected soot	40
Cummins Integral Bypass Trap	Ceramic monolith	Diesel burner heats air over bypassed trap	1
Englehard Catalytic Trap	Ceramic monolith	Catalytic self-regenerating	1

regenerate. Coating a trap with a catalytic material can reduce this regeneration temperature to approximately 800°F. Some regeneration systems divert exhaust from the trap and pass a lower volume of heated air over the filter. Figure 3-12 shows such a system. The filters may be catalytically coated to enhance regeneration. Regeneration is also possible without an external heat source; however, the exhaust temperature must be sufficiently high to allow the particulate to combust. Four-stroke engines are better suited for this approach.

Tables 3-7 and 3-8 shows the emissions of diesel engines with and without particulate traps. The data in Table 3-7 is from a chassis dynamometer over a steady-state cycle, while Table 3-8 shows engine data over the transient FTP. Without taking into account durability, the particulate removal efficiency is well over 75 percent.

There are currently 1,142 buses with particulate traps in field tests throughout the country to assess trap durability. Those with at least 20,000 miles of operation as shown in Table 3-9 (Reference 10). These buses have accumulated over 37.4 million miles in revenue service. From

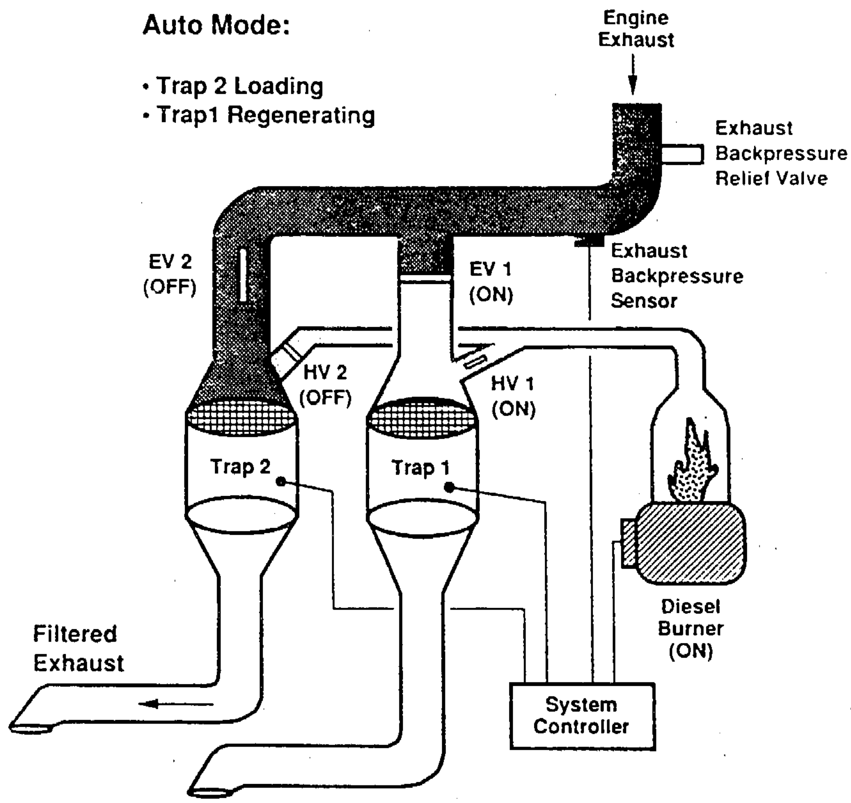


Figure 3-12. ORTECH/Donaldson trap with burner regeneration

Table 3-7. Emissions from RTS transit buses, DDC 6V-92TA engine, 8-mode chassis test

Technology	g/bhp-hr				Particulate Reduction (%)
	NO_x	CO	HC	PM	
MUI[a] Diesel Baseline (240 hp)	6.5	1.7	0.43	0.35	—
MUI Diesel with Donaldson Trap	6.5	0.3	0.09	0.06	83
MUI Diesel Baseline (240 hp)	7.3	2.1	0.51	0.32	—
MUI Diesel with Asahi Trap	6.8	3.1	0.43	0.04	88
DDEC[b] Diesel (277 hp)	9.7	0.7	0.50	0.18	—

[a]Mechanical Unit Injectors.
[b]Detroit Diesel Electronic Control.

Source: Air Resources Board

Table 3-8. Emissions from DDC 6V-92TA engines transient FTP

Engine	g/bhp-hr				Particulate Reduction (%)	Data Source
	NO_x	CO	HC	PM		
Baseline with Elevated Back Pressure	5.01	2.54	0.70	0.350	—	EPA
Donaldson Trap	4.83	3.02	0.59	0.053	85	EPA
Catalyzed Donaldson Trap	4.44	1.62	0.27	0.049	86	EPA
Baseline	4.61	1.63	0.53	0.330	—	DDC
Donaldson Trap	4.80	2.07	0.41	0.060	82	DDC

Table 3-9. Trap equipped buses in service

Location	Bus Manufacturer	Quantity	Fleet Miles
Phoenix, AR	TMC	1	33,000
	MAN	2	85,000
Chatsworth, CA	Neoplan	11	330,000
L.A. County, CA	Neoplan	15	262,000
Los Angeles, CA	TMC	10	695,000
Orange County, CA	Gillig	2	52,000
	TMC	1	23,000
San Diego, CA	Gillig	80	6,000,000
Santa Monica, CA	MCI	10	350,000
Torrence, CA	Gillig	2	96,000
Van Nuys, CA	Neoplan	27	891,000
Minneapolis, MN	Gillig	5	125,000
Albany, NY	OBI	2	60,000
New York, NY	TMC	398	23,084,000
Dayton, OH	TMC	60	3,995,000
Beaumont, TX	TMC	5	300,000
El Paso, TX	TMC	3	150,000
Milwaukee, WI	Neoplan	1	24,000
CANADA	MCI	2	50,000
	TMC	21	840,000
Total		658	37,445,000

As of December 1992

the New York City Transit Authority (NYCTA) experience, DDC and Donaldson have made regeneration system component and software refinements, increasing miles between component failures by a factor of four. NYCTA plans to retrofit 1,998 existing buses with trap oxidizers over the next three years. The DDC 6V-92TA engine and the Cummins L10 engine were certified in 1992 and 1993 with the Donaldson dual trap system. Both engines meet the 1991 California Urban Bus standards. Recent testing at Southwest Research Institute (SwRI) has produced emission levels of 4.8 g/bhp-hr NO_x, 0.4 g/bhp-hr HC, 2.3 g/bhp-hr CO and 0.03 g/bhp-hr PM over the FTP transient heavy-duty diesel cycle for the DDC 6V-92. These engines will be approximately 1.8 times as expensive as the current 1991 counterparts and suffer a 1.3 percent penalty in fuel economy. In a typical urban bus duty-cycle, regeneration will be required every three to six hours (Reference 11).

Donaldson has found that the mean engine hours between incidents (MEHBI) on the particulate traps demonstrated on the NYCTA buses to be currently 8,570 hours. Donaldson expects to increase this to approximately 20,370 hours with new controller software and other modifications currently in use on their two certified engine/trap packages (Cummins L10 and DDC 6V-92). These new system enhancements include a new compact controller package with enhanced hardware and software, improved diagnostic capability interfaced via a single diagnostics unit, and a simplified electrical system. Average speed of the 398 trap-equipped buses at NYCTA is 7.3 miles per engine hour (including idle time). Incidents are defined as any unscheduled trap related occurrence; most of which are minor in terms of their direct cost. The most costly incident is core failure. Secondary core failure (cores which have failed due to undetected failures of other components) are currently a significant portion of the incidents. Primary core failures (cores which have failed without help from other trap system components) are presently one-fifth of the total number of recorded incidents. On certified engines, the entire particulate trap system is covered by five-year, 100,000 mile emissions defect warranty. This is extended to 150,000 miles for the filter cores alone. During this extended period, the user is responsible for the change out labor (3 to 5 hours per core depending upon application) and 50 percent of the replacement core costs. Cores typically cost approximately $1,500 apiece. Donaldson is currently pursuing a core reconditioning/exchange program which will reduce the cost of the core to $500. This program is expected to be available in 12 to 18 months (Reference 12).

3.2.5.2 Oxidation Catalysts

Diesel oxidation catalysts eliminate unburned gases and SOFs in the exhaust. Depending on the type of engine and its exhaust temperature and composition, oxidation catalysts oxidize 30 to 80 percent of the gaseous HC and 40 to 90 percent of the CO present (Reference 13). They do not alter NO_x emissions. Oxidation catalysts have little effect on dry soot (carbon), but engine tests show that they typically remove 30 to 50 percent of the total particulate emissions depending on the engine particulate make-up. This is achieved by oxidizing 50 to 80 percent of the SOFs present. Oxidation catalysts are less effective with "dry" engines in which particulates have very low SOF content.

The main problem with oxidation catalyst application is formation of sulfuric acid and sulfate by oxidation of sulfur dioxide present in the engine exhaust, with consequently large increases in particulate mass emissions. The source of this is sulfur present in diesel fuel; levels as low as 0.05 percent (by mass) can give significant sulfate formation. One approach to this problem is to lower fuel sulfur from the 1994 level of 0.05 to 0.01 percent or lower. A second approach is to develop oxidation catalysts that oxidize HC and CO but not sulfur dioxide. Newly developed catalysts that make almost no sulfate at temperatures as high as 400°C have brought this later step closer to reality. A third approach is to optimize catalyst placement. Sulfur dioxide oxidizes above 350 to 400°C in an oxidation catalyst, while HCs do so below this temperature. Placing the oxidation catalyst so that inlet temperatures favor HC conversion could be a solution (Reference 14).

Since it is technically impossible to make a catalyst completely sulfur-resistant, it is possible as cited above to make a catalyst somewhat resistant. This is only possible by holding exhaust temperatures within a certain temperature range. However, an increase of only 10°C above that range can make the selectivity for the unwanted sulfur reactions start to take over. Diesel engines have too wide a temperature swing for it ever to be possible to make a truly sulfur resistant catalyst.

Reducing the sulfur content of the fuel is a much more practical solution. Going to 0.05 percent sulfur will cost about 3 to 4 cents/gallon of diesel fuel. If oil companies make their pressure vessels (needed to get 0.05 percent sulfur) able to withstand pressures for 0.01 percent sulfur, the reduction from 0.05 to 0.01 percent should cost mere fractions of a cent per gallon.

Oxidation catalysts for diesels must also withstand non-combustible additives in the lubricating oil. Oil might contain zinc, phosphorus, antimony, calcium, magnesium and other contaminants which can shorten catalyst life. Catalyst life can be extended by the development of low-ash lubricating oils and by modifying the carrier properties such as surface chemistry, pore structure and surface area to produce contamination-resistant catalysts.

Some diesel manufacturers favor using oxidation catalysts to reduce particulate and HC emissions over major engine modifications. Oxidation catalysts only add approximately 10 percent to the cost of the engine versus significant costs for engine retooling or particulate traps. Oxidation catalysts provide a cost effective means for light and medium heavy-duty engines to meet a low particulate standard.

3.2.6 1994 Technology Summary

With concern over cost, efficiency, durability and maintainability, engine manufacturers are looking at ways to meet 1994 emission standards without the use of particulate traps. To date, many engine manufacturers believe they can do just that. In fact, many believe they will not even need catalytic converters. In a polling by Fleet Owner Magazine (Reference 15) and more recent discussions with manufacturers directly, the major manufacturers discussed their plans to meet 1994 emission standards. Most will be able to meet these standards without traps, many without catalysts and even some with mechanical fuel injection. A summary of comments is listed below.

Caterpillar suggested that their 10L 3306 engine will stay as a mechanical system, but will have changes in air system, piston and cylinder design and fuel system. Their heavy-duty 3406B will be certified with both an electronic (PEEC) and mechanical fuel injection system, but will probably not require a catalytic converter. The 3176 10L engine, which was first introduced in 1988, will meet the 1994 standards with only minor modifications and will not have a catalytic convertor. Finally, the 3116 6.6L medium-duty engine will require only minor air-system and combustion chamber modifications for 1994, but will have a catalytic convertor. Overall, Caterpillar is unsure about fuel economy on these cleaner engines, but is aiming to improve it over the 1991 versions. By avoiding traps and catalysts, Caterpillar believes that the 1994 engines will not be significantly more expensive than the current 1991 engines, but those with catalysts will have increased costs.

Cummins redesigned their NTC 14L engine in 1990, adding air-to-air aftercooling and integrated electronic controls with unit injectors, as well as higher compression ratios. Cummins currently believes that both the NTC 14 and the L10 engines will not require the use of aftertreatment devices. These engines do require higher pressure injection, more precise control of injection timing, closer turbocharger matching, and other combustion chamber modifications. In their mid range B and C series engines, Cummins will use catalytic converters to control hydrocarbon and particulate emissions. Even without traps and catalysts, Cummins warns that the cost to develop these engines is significant and therefore the 1994 engines will have a higher cost than the 1991 versions.

Detroit Diesel Corporation (DDC) believes they can meet the 1994 emission standards with the 11.1L and 12.7L versions of their Series 60 engine without any type of aftertreatment. By increasing injection pressure, they have already brought the Series 60 down to 0.08 g/bhp-hr PM. DDC believes they also may have some minor changes in turbocharger design, but not much more than that. They expect to maintain current fuel economy performance without any dramatic increases in cost. DDC will certify their 2-stroke 71 and 92 Series engines using credits, as these engines are mostly used in urban bus applications. Urban buses must meet a 0.07 g/bhp-hr PM standard in 1994. The Series 92 will also be available with an active particulate trap. DDC expects to meet the 1994 bus standard with their Series 50 engine using high injection pressure, a turbocharger with a ceramic coated turbine wheel and an exhaust catalyst.

General Motors (GM) only diesel engine that will be certified for 1994 is the newly redesigned light-duty 6.5L V-8. This new engine, which is based upon the naturally aspirated 6.2L diesel, was redesigned to include turbocharging, a reworked indirect-injection fuel system and the Ricardo Comet swirl chamber. GM plans to use a catalytic convertor in 1994, but to stay with the mechanical fuel injection system and use no aftercooling. This engine utilizes a high swirl, moderate fuel injection pressure scheme which is more cost effective in smaller engines, but the indirect injection results in a 15 percent fuel consumption penalty in comparison to direct injection engines.

Navistar showed in 1989 that they could meet 1994 emission standards with their prototype 466 "Smokeless Diesel" engine with just a passive catalytic converter and lower sulfur diesel fuel. Combustion chamber improvements alone could bring both the 466 and 360 medium-duty engines within 1994 limits. These would include injection pressure increases, improved injector locations, reduced ring land area, tighter oil control and better air management. They have decided to stick with their current Bosch mechanical fuel injection system on both engines. Finally, they suggest that the light-duty indirect-injection 7.3L engine will probably require a catalyst for 1994. Navistar adds that this may be a more cost effective measure even for its other engines.

From the statements above, apparently most manufacturers can provide low particulate emissions with some engine redesign. Most manufacturers have ruled out using particulate traps and many resist using catalytic converters. It also is apparent that 0.07 g/bhp-hr can be reached in these engines using either traps or catalysts, even if NO_x emissions need to be lowered through injection timing changes. Meeting greatly reduced NO_x emissions and still meeting 0.07 g/bhp-hr, however, is another issue altogether.

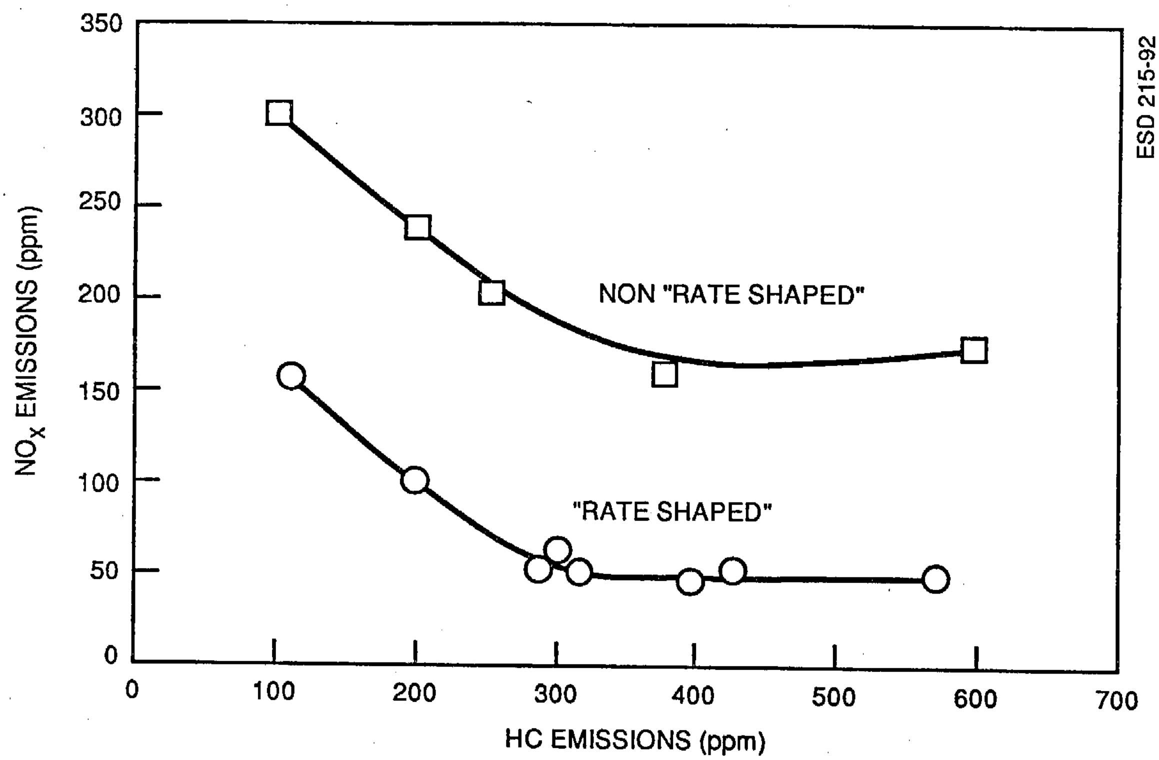
3.3 FUTURE ENGINE AND AFTERTREATMENT TECHNOLOGY

As discussed above, engine manufacturers will be able to meet the 1994 emission standards with most of their current engine lines. With the event of the 1991/1994 emission standards, manufacturers phased out some engine lines, redesigned others and created new designs. This process will continue as emission standards become more strict. In addition, as NO_x and PM standards were lowered, substantial problems in engine-to-engine variability, oil sludging and cold start operation needed to be solved.

With substantial advancements in fuel, air and combustion systems and with the use of advanced aftertreatment, diesel engines might be able to reach Scenario 1 goals of 2 g/bhp-hr NO_x and 0.05 g/bhp-hr PM. Engine modifications needed to reach these emission levels will include advanced fuel injection systems, advanced turbochargers and charge air cooling, exhaust gas recirculation, and advanced diesel fuel formulations. Advanced aftertreatment includes catalytic particulate traps with sophisticated regeneration systems, highly efficient oxidation catalysts and lean NO_x catalysts. While research and testing of these advanced concepts is currently underway, support and resources will determine not only the outcome of these projects, but the timeframe for development and production. A description of these research areas and progress to date are discussed in the following subsections.

3.3.1 Advanced Fuel Injection

The rate at which fuel is injected into the engine cylinder can affect both NO_x and PM emissions. By varying the rate at which fuel is injected into the cylinder during the injection period, some further improvements in emissions can be realized. Varying the fuel injection rate during the initial injection is called "rate shaping." Improvements in emissions are realized by injecting only a small quantity during the first phase of injection. Rate shaping reduces the ferocity of the initial detonation due to reduced fuel in the cylinder. This provides for smoother combustion, lower combustion temperatures, less combustion noise and reduced mechanical stress. Since the initial fuel injected into the cylinder is limited, formation of NO_x emissions is reduced. Rate shaping is particularly advantageous at low loads and speeds as shown in Figure 3-13 (Reference 16). Navistar has developed a hydraulically actuated, electronically controlled unit injector (HEUI) system that allows for programmable injection characteristics and high injection pressure at all loads and speeds.



Source : Navistar Engines

Figure 3-13. Effect of rate shaping on NO_x—low idle

Using a lube oil actuated system, fuel injection pressures reach a maximum of 22,000 psi. Navistar claims to have problems in the variability of their rate shaping, and that significant development time will be required to perfect their system for production.

Ultra high fuel injection pressures have been examined by the Advanced Combustion Engineering Institute in Japan (Reference 17). This system uses injection pressures of 35,000 psi and shows dramatic improvements in particulate emissions at steady state conditions. In addition, NO_x emissions can be reduced by extending the lean limit. Unfortunately, this system requires a completely quiescent chamber with virtually no air motion to keep the injection plume from breaking up. In addition, the injection pump needed for the ultra high pressures is extremely large and not currently practical for mobile engines.

3.3.2 Advanced Turbocharging and Charge Cooling

Turbocharger matching will be an important issue in future engines. Two types of advanced turbochargers show significant improvements in fuel consumption and emissions. These are the variable geometry turbocharger and the turbo expander.

Variable geometry turbochargers (VGT) provide leaner air/fuel ratios under full load conditions. By varying the geometry of the turbocharger, quick response at low speeds is achieved without causing excessive boost at high speeds. With advanced control techniques, a VGT can be optimized to give a significant reduction in particulate emissions through improved transient response, and might be expected to achieve up to 15 percent improvement in particulate emissions at constant NO_x with a 2 to 3 percent improvement in BSFC (Reference 18). In addition, a VGT will be of use for giving the correct pressure conditions across an engine for EGR flow without an additional pump.

Turbo expanders over-compress the intake charge, cool it by use of an aftercooler, then expand it down to a lower pressure. The expansion further reduces the intake temperature which greatly reduces both particulate and NO_x emissions and improves BSFC. The major disadvantage is the high cost of the system which may limit application to very few premium engines. In addition, pumping losses are significant, but might be offset by the need for reduced injection timing retard, thus giving better fuel consumption.

Proper design of air-to-air aftercoolers will also be important to keep the intake air temperature low to minimize NO_x emissions.

3.3.3 Advanced Diesel Fuel

In addition to reducing the aromatic and sulfur content of fuels, fuel additives can show significant improvements in emissions. Two of the most promising additives are cetane improvers and oxygenates. Cetane improvers increase the cetane number of the fuel and shorten the ignition delay period. Oxygenates add oxygen to the fuel, thereby leaning the fuel/air mixture and resulting in lower particulate emissions.

Cetane improvers can provide substantial reductions in emissions. By increasing cetane number from 40 to 53, NO_x emissions are reduced 9 percent, HC 42 percent, CO 56 percent and PM 10 percent (Reference 8).

Oxygenates such as water, alcohols, esters, di-esters, gylcols, ethers, glycol ethers, nitrates, di-nitrates and di-carbonates add bound oxygen to the fuel and can significantly reduce PM emissions. In choosing an oxygenate, one must examine volatility, solubility, toxicity, odor, seal compatibility and cost, in addition to possible side effects. In a study done by DDC (Reference 19), diglyme (diethylene glycol dimethyl ether) was chosen since it is relatively safe and contains 36 percent oxygen. In a 6V-92TA engine, diesel fuel with 5 percent diglyme added reduced particulate emissions 20 percent over diesel fuel with no oxygenate added.

Chevron is currently selling a reformulated special diesel fuel with a cetane number of 52, sulfur at 350 ppm and aromatics at less than 30 percent. While this exceeds California's 1993 specification for aromatics, Chevron claims up to a 40 percent reduction in HC, 20 percent reduction in CO, 9 percent reduction in PM and 2 percent reduction in NO_x in most HD engines when compared to current diesel fuels. This fuel was introduced into the South Coast Basin in June 1990. Chevron is currently formulating a cleaner diesel fuel to meet the 1993 requirements. Other reformulated diesel fuels will also be available in the October 1993 timeframe.

3.3.4 Advanced Particulate Traps

Future particulate traps are expected to have reduced complexity, fewer components and lower cost. At present, particulate traps are expensive (more expensive than medium-duty engines), need improved durability and require complex controls for regeneration. Current development efforts are to redesign trap systems for lower cost, reduced power consumption, and improved reliability. One new developmental system involves electrical generation with 12 volt operation, using exhaust oxygen along with poppet valves for each of 4 filters and microprocessor control. This configuration is currently being tested by Donaldson Co. on a Dodge pickup truck with a Cummins 6BT 5.9L 160hp engine (Reference 20). Currently available commercial trap systems are also being further optimized to provide significantly improved performance at lower costs.

3.3.5 Advanced Catalytic Aftertreatment

Two types of catalysts are currently emerging as possible future aftertreatment technology. These are the advanced oxidation catalyst and the lean NO_x catalyst. The advanced oxidation catalyst will oxidize PM, CO and HC emissions without increasing sulfate emissions. The lean NO_x catalyst uses reductants such as urea, ammonia or hydrocarbons in diesel exhaust to catalytically reduce NO_x emissions.

Oxidation catalysts currently oxidize 50 percent of SOFs, 30 percent of total PM and 30 percent of gaseous HC emissions. When sulfur is present in the fuel, they also manufacture sulfates. Future oxidation catalysts need to be designed to reduce SOF 60 to 70 percent, reduce total PM emissions 20 to 40 percent and gaseous HC emissions 50 to 60 percent. In addition, future oxidation catalysts need to minimize the creation of sulfates, or use very low sulfur fuels (<100 ppm). Significant breakthroughs in noble metals and catalyst washcoat will bring these emission reduction goals within sight. Durability and cost are still issues, but oxidation catalysts can complement new engine technology by allowing manufacturers to concern themselves with NO_x reduction and deal with excess particulates through exhaust aftertreatment.

Lean NO_x catalysts offer another approach by catalytically reducing NO_x emissions to nitrogen. As diesel engines operate lean and have excess air, current technology NO_x catalysts, used in light-duty automotive applications, cannot be used. Lean NO_x catalysts use zeolite catalysts and a reducing agent to reduce NO_x. In selective catalytic reduction (SCR) systems, ammonia (NH_3) or urea is used as a reducing agent. For steady-state applications, NO_x reductions of up to 90 percent have been reported (Reference 21), but this process is difficult for use in vehicle applications. In vehicle applications, a 25 percent efficiency would be expected. Such systems might be available in 10 years.

Recently, a great deal of attention has been given to DE-NO_x catalysts (Diesel Engine NO_x catalysts designed to reduce NO_x emissions in fuel lean environments) that use exhaust hydrocarbons as the reducing agent. By using copper zeolite sieves, large molecule hydrocarbon species are trapped within the catalyst during periods of high hydrocarbon emissions (idle and low load), and then react with NO_x during periods of high NO_x emissions (high loads). The catalyst operates most effectively in the 350 to 600°C temperature range. Since exhaust temperatures rise with a drop in air-fuel ratio (A/F), catalyst efficiency is inversely proportional to A/F. Most diesel engines operate at A/F greater than 22 to limit smoke and PM emissions. In order to raise temperatures sufficiently for catalyst operation, the A/F would need to drop in the range of 16 to 18, with a potential NO_x reduction of 70 to 50 percent, respectively. This would increase fuel consumption 20 to 10 percent, respectively. Navistar claims that these operating conditions can be met with their hydraulically-activated EUI (HEUI) system without increased smoke, however, Ricardo's experience shows that there would be a significant increase in soot particulate emissions at these reduced A/Fs, even with high pressure injection. Fuel injection may need to be modified to provide additional HCs in the exhaust system to act as a reductant. Specific zeolites may also need to be engineered to capture different classes of hydrocarbons found in diesel exhaust. Similar efforts could be applied to methanol, NG, and LPG exhaust.

SwRI has formed a consortium to develop such a DE-NO_x catalyst system. SwRI estimates that they will have a research catalyst solution which reduces NO_x emissions by 50 percent in the next three to six years (Reference 22). Present work indicates that these catalysts are highly sensitive to HC, NO_x, SO_2 and water concentrations in the exhaust, as well as temperature and space velocity as shown in Figure 3-14 (Reference 3). At present, these catalysts are only 20 percent efficient and must be specifically tailored to each engine. Development time on these catalysts for production will be greatly extended unless concentrated research is done.

Further research on DE-NO_x catalysts in Japan has shown that the addition of diesel fuel to the exhaust in front of a DE-NO_x catalyst in concentrations of 6000 to 8000 ppm carbon will

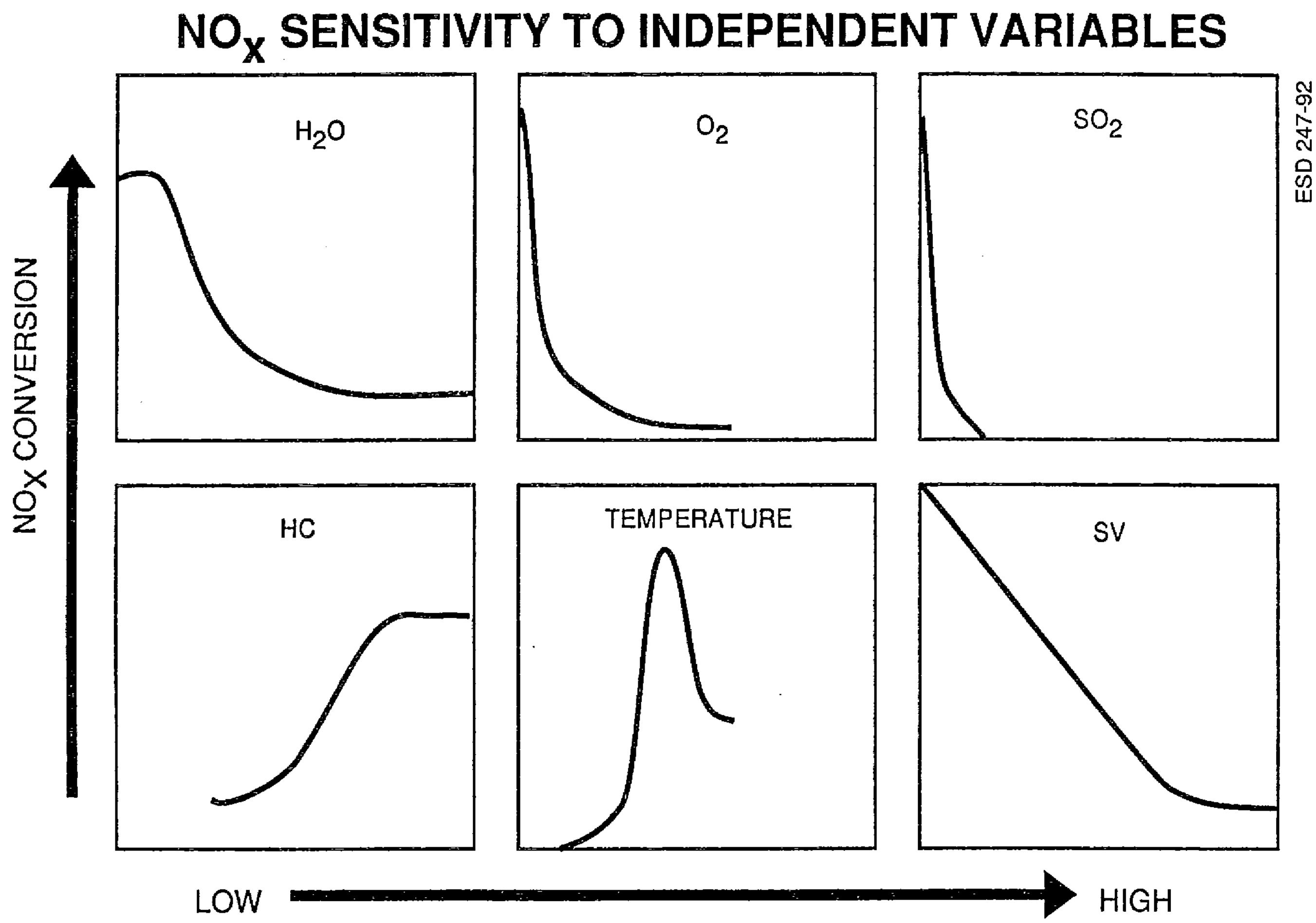


Figure 3-14. Effects of various parameters on DE-NO_x catalyst efficiency

increase NO_x reduction efficiency to nearly 80 percent at specific conditions (Reference 23). These amounts of fuel correspond to approximately 5 percent of the fuel consumption, which is significantly less than the 10 to 20 percent needed for the SwRI system.

3.3.6 Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) is one of the most effective methods for reducing NO_x emissions to the 2 g/bhp-hr scenario goals without fuel consumption penalties. This is due to the reduction in peak cylinder temperatures through dilution of the charge with combustion residuals of high specific heat capacity. Two research organizations, Ricardo and SwRI, are studying the use of EGR.

The major difficulties with EGR are increased smoke and particulate emissions, partly because of reduction in net air/fuel ratio, and partly because of poorer combustion efficiency. Also, severe engine wear problems have resulted from lubricating oil contamination by carbon and sulfuric acid created from fuel sulfur (Reference 24). On the other hand, at light load operating conditions, EGR can have a very beneficial effect on emissions, giving HC and NO_x reductions through shorter ignition delay and charge dilution.

Ricardo, SwRI and some major OEMs consider EGR to be one of the primary NO_x control technologies for the future. However, extensive development needs to be performed in the following areas:

- The combustion system must be made tolerant to EGR to accomplish low carbon emissions at part load and full load
- Recirculation and admission systems must avoid unfavorable pressure gradients and give adequate flow
- EGR cooling should be used for high load operation at very low NO_x levels. At part load, hot EGR will probably be acceptable. Control and cooler sizing/cost will be key issues

- Turbomachinery will need to be rematched for increased boost at full load. Computer simulation has shown that a VGT will give acceptable EGR flow and increased boost. A wastegated fixed geometry turbocharger might be a less efficient alternative.
- Wear, fouling and durability issues will involve development of lubricating oil specification and conditioning systems, and the mechanical design of the engine
- Transient control strategy will need to be developed

Ideally, EGR would be combined with the combustion and fuel injection system developments mentioned previously, since particulate control is a fundamental part of successful EGR applications.

Ricardo Consulting Engineers are developing EGR strategies for obtaining low NO_x emissions, as shown in Figure 3-15 (Reference 25). These utilize hot EGR at low load to improve ignition delay, thereby reducing particulate and HC emissions as well as NO_x. Towards full load, cold EGR is used to reduce combustion temperatures and thus NO_x emissions. Since EGR applied at full load tends to increase smoke emissions due to operation at lower air/fuel ratios, turbocharger boost pressure is increased to reduce smoke emissions. At low loads, the fuel injection rate is increased to prevent flame-out of overly lean mixtures, thereby minimizing smoke emissions. Use of this strategy can obtain low NO_x emissions without loss in fuel economy. Ricardo believes this EGR strategy will result in 2.0 g/bhp-hr NO_x, but particulate emissions will rise to around 0.15 g/bhp-hr, requiring the use of a particulate trap to obtain 0.05 g/bhp-hr PM. While use of EGR is still in the research stage, results to date give cause for optimism.

3.3.7 Advanced Combustion Systems

A most promising area of research is detailed development of combustion chamber profile. The combustion chamber needs to be designed to optimize air utilization and burning characteristics for a given injection pressure/swirl/EGR combination. The key issue will be to redefine the optimum geometry when optimizing the combustion system with EGR. This could result in some reduction in PM emissions over a wide variety of engine operating conditions.

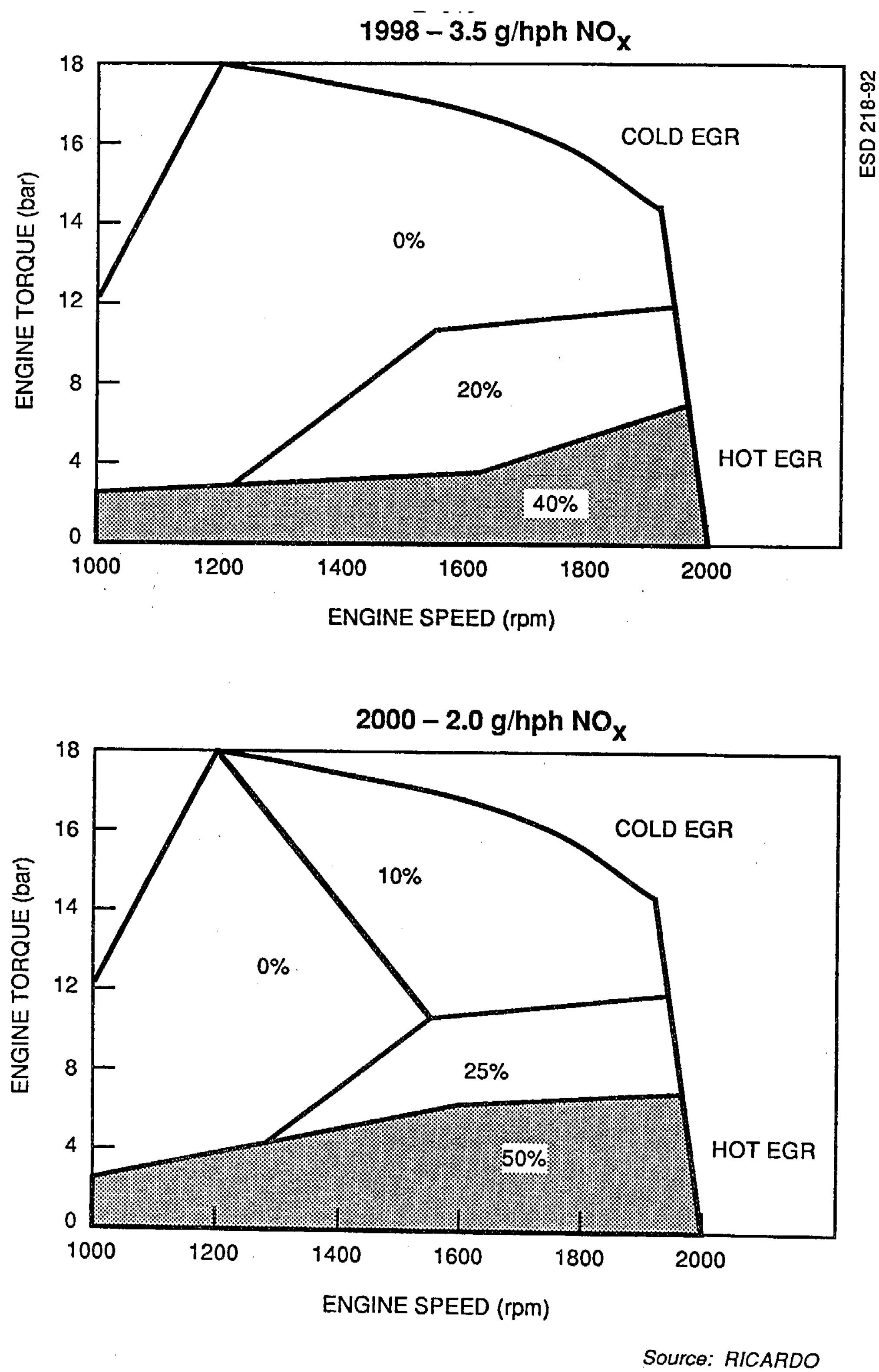


Figure 3-15. Possible EGR strategies

Low heat rejection diesel engines are another topic of substantial research. By minimizing heat loss through use of ceramic pistons and cylinder liners, fuel consumption and particulate emissions are substantially reduced. However, full combustion chamber insulation can cause NO_x emissions to increase as much as 50 percent, necessitating substantial injection timing retard. At this time, low heat rejection diesel engines do not appear consistent with low NO_x emission strategies.

3.3.8 Advanced Electronic Control

Electronic control will play a very important role in future diesel-fueled engines. In addition to injection timing and duration being electronically controlled, turbocharging, EGR flow rate, aftercooler bypass and particulate trap regeneration may also be controlled.

Transient timing control could be expected to improve particulate emissions by 5 to 10 percent by overcoming increased ignition delay and other effects that influence instantaneous emissions during the FTP. This represents a new technology which might make use of advanced "predictive" or "feed-forward" techniques. Such techniques could also improve the transient response of an advanced EGR/turbocharger system.

Two exhaust sensors are being examined for closed-loop feedback control to provide better optimization of engine operating variables. These are the lean oxygen sensor, and the EUGO or NO_x sensor. The lean oxygen sensor is currently available and provides a linear signal based upon air/fuel ratio in the lean region. This can be used to control engine conditions so that the air/fuel mixture does not become too lean. However, feedback control systems will have to be improved to respond fast enough for transients during an FTP cycle.

The EUGO or NO_x sensor detects the amount of NO_x in the exhaust. By using this sensor, closed-loop feedback control of EGR rate could be optimized for each engine condition for a low NO_x engine. Navistar is investigating these sensors for use in their future engines. At present only prototypes of these sensors exist and will require substantial improvement in transient response to be useful in closed-loop control.

3.3.9 Future Technology Summary

Based upon the above information, it seems possible to reach the Scenario 1 goals of 2.0 g/bhp-hr NO_x and 0.05 g/bhp-hr PM at a research laboratory level. However, much research and development will be needed in the next few years to make this a reality for mass production with acceptable durability, cost and deterioration factors.

Two paths of development are currently being sought to bring diesel engines within the Scenario 1 goals. These are engine modifications including EGR, and use of DE-NO_x catalysts. Both technologies require substantial development time, but will probably be available in the next 7 to 10 years. It is estimated that future engines capable of meeting Scenario 1 goals will cost over 2 times as much as their 1994 counterpart.

In Figure 3-16, Ricardo outlines a possible development path to reach Scenario 1 goals. Through the application of EGR, the 1994 clean diesel engine will produce 2.0 g/bhp-hr NO_x, but at 0.25 g/bhp-hr PM. By optimizing the combustion, air and fuel systems with EGR, Ricardo estimates that the particulate emissions can be reduced to 0.15 g/bhp-hr PM at 2.0 g/bhp-hr NO_x. Finally by using a particulate trap, the future diesel engine particulate emissions can be reduced to 0.05 g/bhp-hr PM. These emission levels can be attained without a significant increase in HC emissions or BSFC. With light EGR strategies, Ricardo believes that 3.5 g/bhp-hr NO_x and 0.10 g/bhp-hr PM can be reached.

Advanced aftertreatment will also play an important role in future diesel engines. Oxidation catalysts and DE-NO_x catalysts offer solutions to lower cost engines. Particulate traps will probably be needed to reach low PM standards at low NO_x levels. The estimated effectiveness of exhaust aftertreatment devices is given in Table 3-10.

In summary, the advanced diesel engine will most likely involve EGR, NO_x catalysts, oxidations catalysts and/or particulate traps. Beginning with a 1994 technology "Base" engine, the effect of various emission reduction strategies on emissions, power and fuel consumption are shown in Table 3-11. These technologies are described below.

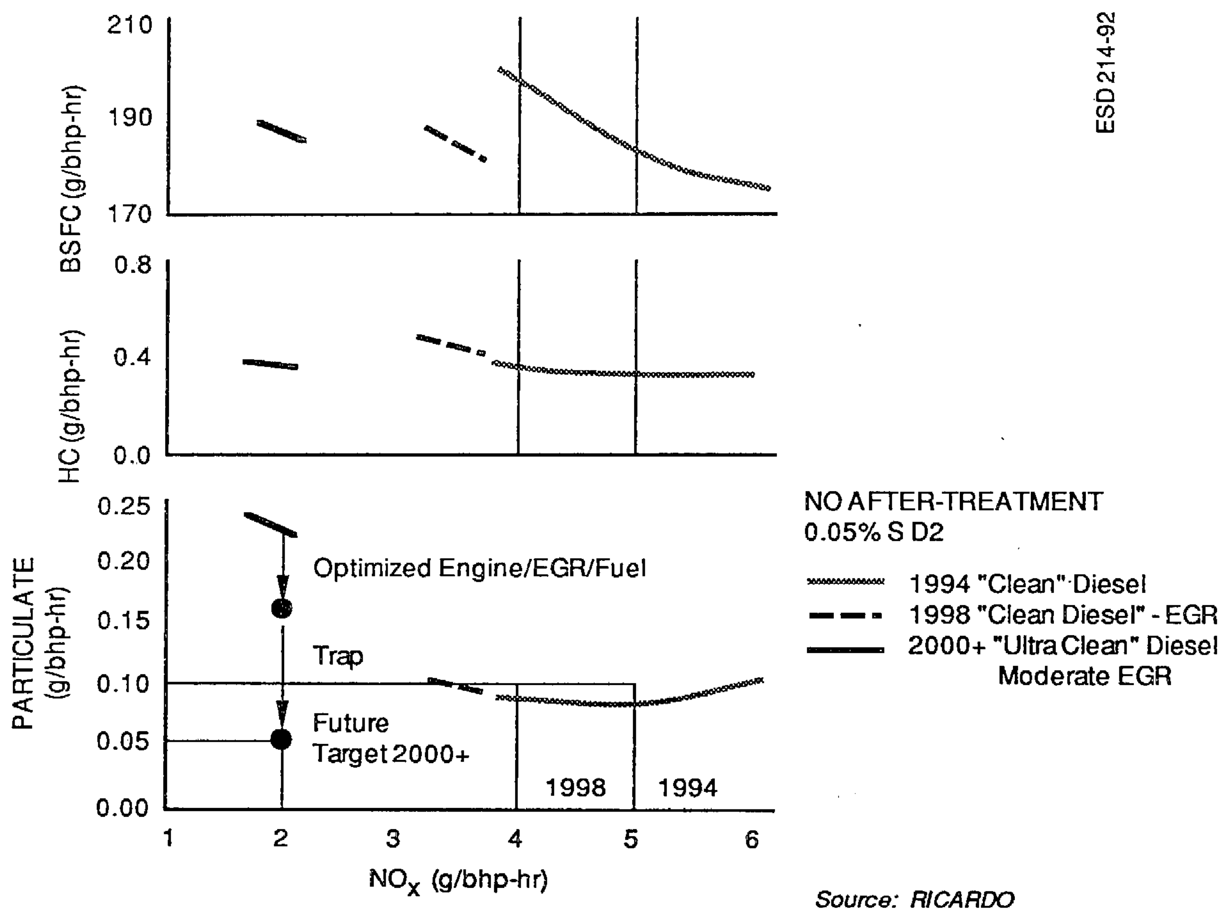


Figure 3-16. Development steps for future 2000+ "ultra clean" diesel (U.S. FTP results—g/bhp-hr)

Table 3-10. Exhaust aftertreatment technology efficiencies (% reductions)

	Particulate Traps (%)	Oxidation Catalyst (%)	DE-NOx Catalysts (%)
PM	50 to 90	20 to 40	0 to 10
NO_x	0	0	30 to 50
HC	9	20 to 90	20 to 90
CO	12 to 15	20 to 90	20 to 90

Table 3-11. Expected emissions of base engine with additional technologies

	Base	+EGR	+EGR/Trap	+EGR/OxCat	+SCR	+DE-NO_x	+DE-NO_x/EGR
NO_x (g/bhp-hr)	4.0	<2.0	<2.0	<2.0	1.5-2.0	2.0	1.5
Particulate (g/bhp-hr)	<0.1	>0.2	<0.05	0.15	0.15-0.2	0.09	0.15
Gaseous HC (g/bhp-hr)	<0.3	<0.3	<0.3	0.1	0.1	<0.3	<0.5
BSFC (g/bhp-hr)	190	190	195	190	195	205	215
Rated BMEP (psi)	200-230	185-220	185-220	185-220	185-220	200-230	185-220
Peak BMEP (psi)	275-335	260-305	260-305	260-305	260-305	275-335	260-305
Possibility	very good	good	moderate	good	poor	moderate	moderate

Exhaust Gas Recirculation

- Cooled EGR applied at high load conditions, modulated cooling to give hot EGR at part load
- Electronic control of VGT and EGR admission system
- Base engine combustion, fuel and air systems optimized to give good EGR tolerance with minimum particulate penalty

NO_x Catalyst

- Selective catalytic reduction of NO_x using reducing agent added to exhaust system. Precise technology not yet finalized
- DE-NO_x catalyst

Oxidation Catalyst

- Selective oxidation of gaseous and particulate HC with metal catalyst. No effect on NO_x

Particulate Trap

- Collection and on-board regeneration of total particulates. No effect on NO_x.

3.4 OVERALL DIESEL TECHNOLOGY SUMMARY

With the advent of 1991 emission standards, engine manufacturers had to eliminate some of their dirtier engine lines. Several manufacturers totally redesigned old lines and others built entirely new engines. These new engines were built with 1994 emission standards in mind. As a result, many manufacturers will be able to achieve 1994 emissions without total engine redesign. Older engines will need to be redesigned or use exhaust aftertreatment to meet the 5 g/bhp-hr NO_x/0.1 g/bhp-hr PM 1994 standard.

As future emission regulations are passed that significantly reduce NO_x and PM emissions, engine manufacturers will again have to redesign, scrap and rebuild engines for the new standards. To meet low NO_x and PM emission goals, significant advances in engine technology will need to occur. Several research areas show much promise of reaching fruition in the next 7 to 10 years.

By combining advanced combustion system design with EGR and catalytic particulate traps, Scenario 1 emission levels could be reached. An efficient DE-NO_x catalyst could also be used to reach Scenario 1 emission levels if combined with an advanced oxidation catalyst or particulate trap. In the near term, by utilizing EGR and advanced catalytic trap technologies, manufacturers could produce diesel engines capable of meeting 2.5 g/bhp-hr NO_x and 0.1 g/bhp-hr PM by 2000+ with a price tag of 1.5 to 2.0 times the cost of an equivalently rated 1994 engine. As research and developed efforts continue, engines will be able to meet 2.0 g/bhp-hr NO_x and 0.05 g/bhp-hr PM but at over 2 times the price of the 1994 engine. Substantial research and development efforts are needed during the next several years to produce low emission engines that will have good reliability and durability. Technology forcing regulations and the threat of alternative fuels may well be the incentive to make this happen.

SECTION 4

GASOLINE ENGINE TECHNOLOGIES

Gasoline engines power 70 percent of light heavy-duty vehicles and 44 percent of medium heavy-duty vehicles. The low speed, high torque requirements of heavy heavy-duty vehicles make gasoline engines less desirable.

Heavy-duty gasoline engines are tested with a different test procedure and are regulated by different emission standards than heavy-duty diesel engines. While NO_x emission standards are the same for gasoline and diesel engine test cycles in the heavy-duty engine range, there is no particulate standard for the gasoline engines. The 1991 California standards for gasoline heavy-duty engines are listed in Table 4-1.

4.1 ENGINE TECHNOLOGIES

Current heavy-duty gasoline engines are electronically fuel injected with closed-loop feedback control and an oxygen sensor. These engines run stoichiometric throughout most of the operating range. Catalysts and EGR are used to control emissions. Specifications and emissions for 1992 heavy-duty gasoline engines are listed in Table 4-2.

Table 4-1. 1991 California heavy-duty gasoline truck standards

GVWR	(g/bhp-hr)			
	THC[1]	NMHC[1]	CO	NO_x
8,500 to 14,000	1.1	0.9	14.4	5.0
> 14,000	1.9	1.7	37.1	5.0

[1] Must either meet THC or NMHC standard

Table 4-2. 1992 EPA certification data for heavy-duty gasoline engines[1]

Manufacturer	Engine Model	Cylinders	Displacement (l)	Fuel System	Aspiration	Class	Emission Control System[2]	HP @ Speed	Torque @ Speed	Emissions (g/bhp-hr)		
										HC	CO	NO_x
Chrysler	360-1	V-8	5.9	Fuel Injection	Natural	Light	Air/EGR/Cat/OL	205@4000	305@2400	0.7	12.4	4.3
Ford	5.8L Manual	V-8	5.8	Fuel Injection	Natural	Light	EGR/Air/Cat/CL	210@3800	310@2800	0.4	10.7	3.2
Ford	5.8L Auto	V-8	5.8	Fuel Injection	Natural	Light	EGR/Air/Cat/CL	210@3800	310@2800	0.5	9.4	3.3
Ford	7.0L Manual	V-8	7.0	Fuel Injection	Natural	Medium	EGR/Air/CL	230@3500	360@2800	1.2	17.7	3.5
Ford	7.0L Auto	V-8	7.0	Fuel Injection	Natural	Medium	EGR/Air/CL	235@3500	360@2800	0.8	16.9	4.0
Ford	7.5L Manual	V-8	7.5	Fuel Injection	Natural	Medium	EGR/Air/Cat/CL	218@3600	363@1800	0.5	9.2	3.7
Ford	7.5L Auto	V-8	7.5	Fuel Injection	Natural	Medium	EGR/Air/Cat/CL	218@3500	363@1800	0.5	8.0	4.0
General Motors	LB4	V-6	4.3	Fuel Injection	Natural	Light	EGR/Cat/CL	155@4000	230@2400	0.5	7.5	3.7
General Motors	LO4	V-8	5.7	Fuel Injection	Natural	Light	EGR/Cat/CL	190@4000	300@2400	0.5	9.1	3.5
General Motors	LSO	V-8	6.0	Fuel Injection	Natural	Light	EGR/Cat/CL	210@4000	325@2400	0.8	5.4	4.6
General Motors	LRO	V-8	7.0	Fuel Injection	Natural	Medium	EGR/Cat/CL	240@4000	375@2800	0.9	10.4	3.5
General Motors	LI9	V-8	7.4	Fuel Injection	Natural	Medium	EGR/Cat/CL	230@3600	385@1600	0.6	14.0	3.9

[1]Source: EPA Federal Certification Test Tesults for 1992 Model Year.
[2]Air = air injection.
EGR = exhaust gas recirculation.
Cat = catalyst.
CL = closed loop.
OL = open loop.

Ford Motor Company uses REDOX catalysts on their gasoline heavy-duty engines to control emissions. Ford's REDOX catalysts are single precious metal catalysts containing no rhodium. The reduction efficiency for NO_x is estimated to be 20 to 40 percent. This type of catalyst can withstand much higher temperatures than conventional three-way catalysts and is much less temperature sensitive. Ford has put the catalyst 36-40" downstream from the exhaust manifold flange to avoid damage to the catalyst from excessive exhaust temperatures. In addition, after a predetermined amount of time at full throttle, the air/fuel ratio goes rich to increase power, reducing NO_x emissions and exhaust temperature. An air pump is used to provide enough air to oxidize the engine-out HC and CO emissions (Reference 26).

General Motors (GM) uses oxidation catalysts on all engine models except the 7.4L V-8 engine. The 7.4L engine uses a three-way catalyst with low rhodium content and places it after the muffler. Catalyst temperature was monitored during development testing and, based on that experience, a control strategy was designed to predict when excessive exhaust temperatures could occur and activate fuel enrichment to limit those temperatures. The additional fuel cools the exhaust through fuel vaporization. However, during most of the time, these engines run stoichiometric. No air pump is used on any of GM's heavy-duty gasoline engine lines (Reference 27).

4.2 REFORMULATED GASOLINE

Reformulated gasoline adds another element to cleaner exhaust emissions. California has defined Phase II gasoline, which will be required throughout California in 1996. Phase II gasoline reduces total aromatic hydrocarbons to less than 25 percent by volume, benzene to less than 1 percent by volume, olefins to less than 5 percent by volume, RVP to 7.0 psi and sulfur to less than 40 ppm. In addition, the 50 percent and 90 percent distillation temperatures (T_{50} and T_{90}, respectively) are also regulated. T_{50} is limited to a maximum of 210°F and T_{90} to a maximum of 300°F. The oxygen content in the fuel must be a minimum of 1.8 percent by weight year-round and a maximum of 2.7 percent by weight in the winter. Advances in reformulated gasoline could reduce overall reactivity of gasoline exhaust in the atmosphere by approximately 30 percent over normal gasoline. Reformulated gasoline can be used in existing vehicles without modification and provide significant air quality benefits.

4.3 GASOLINE ENGINE TECHNOLOGY SUMMARY

Much of the research in light-duty gasoline engines can be applied to heavy-duty engines, but emission control technology is hampered in heavy-duty gasoline engines by the high exhaust temperatures at high load conditions. Extensive research and development of new engines and control systems has enabled current Otto-cycle engines to meet 1991 emission standards as shown in Figure 4-1. Further reductions in NO_x are possible with increased EGR and ignition retard at the expense of fuel economy. Engine manufacturers could probably reach 3 g/bhp-hr NO_x with some increase in fuel consumption. Improved REDOX catalysts are currently under development that use a Pt/Pd formulation and can reduce NO_x emissions by 70 percent (Reference 28). Since these catalysts contain no rhodium, they can withstand higher exhaust temperatures and can be mounted nearer to the engine for faster light-off. With significant improvements in catalyst technology and more sophisticated feedback control, gasoline engines may be able to approach 2 g/bhp-hr NO_x.

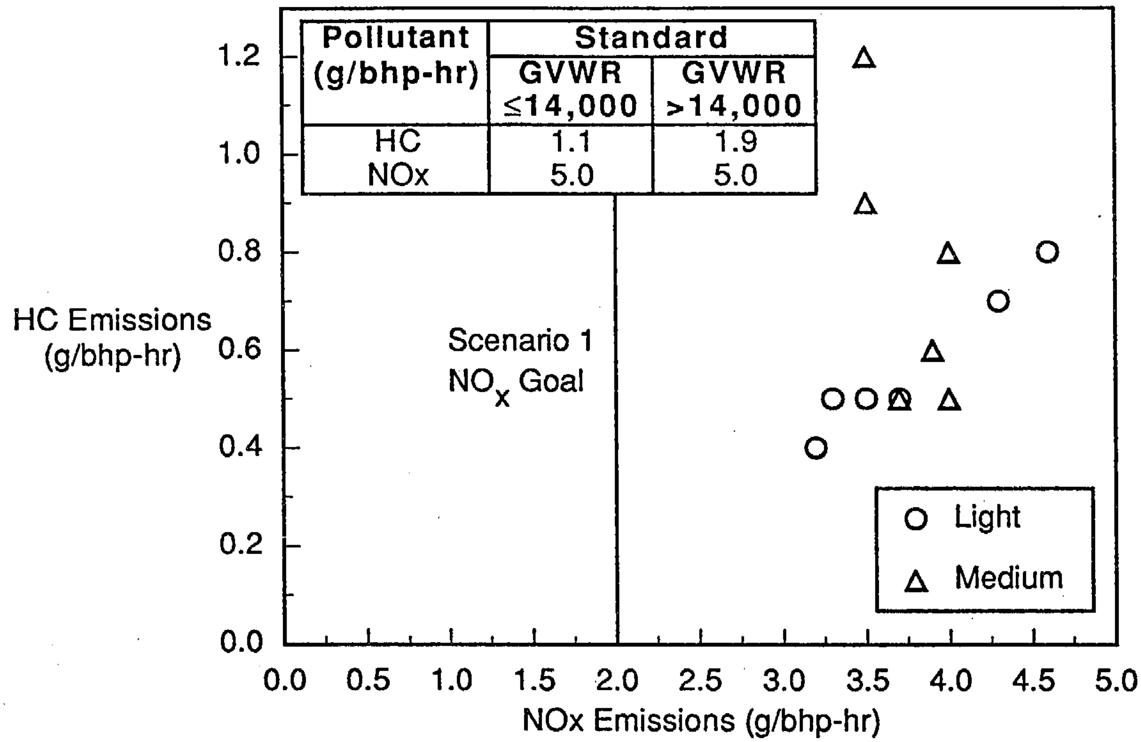


Figure 4-1. EPA HC and NO_x certification emissions for 1992 heavy-duty gasoline engines

SECTION 5

ALCOHOL FUEL TECHNOLOGIES

Alcohol fuels offer great potential for low NO_x and particulate emissions from heavy-duty engines. The chemical characteristics of alcohols provide both advantages and disadvantages compared to diesel fuel, but the disadvantages may be overcome technologically. Table 5-1 shows the properties of the two alcohols considered here, methanol and ethanol, in comparison to diesel fuel.

Alcohol fuels produce low particulate and NO_x emissions levels because of their chemical structure and properties. They burn without producing particulates because of their simple molecular structure and the fact that they have a carbon-oxygen bond which tends to form CO or CO_2 rather than the carbon compounds that constitute particulate emissions. Alcohols also have a relatively high heat of vaporization. Upon injection of alcohol into the cylinder, the charge temperature is cooled due to fuel vaporization, substantially more than an energy equivalent amount of diesel fuel. While this makes the charge more difficult to autoignite, it reduces the peak cylinder temperature, thereby reducing NO_x emissions. The adiabatic flame temperature of alcohols is lower than that of diesel fuel, also lowering combustion temperatures. Lower combustion temperatures produce less NO_x.

However, many of these same chemical properties make alcohols difficult to ignite. Alcohols' higher heat of vaporization and lower energy density, cause the compressed charge temperature to drop rapidly upon injection of the fuel due to vaporization. This substantially slows autoignition reactions and lengthens ignition delay. If the ignition delay becomes too long, the charge will not burn completely resulting in high HC and CO emissions. Partial oxidation of

Table 5-1. **Properties of methanol, ethanol, and diesel fuel**

Properties		Methanol	Ethanol	Diesel Fuel #2
Chemical Formula		CH_3OH	CH_3CH_2OH	$C_nH_{1.8n}$ (C_8 to C_{20})
Molecular weight		32.04	46.07	≈170
Element (wt %)	C	37.49	52.14	86.88
	H	12.58	13.13	13.12
	O	49.93	34.73	0
Density (@ 60°F, 1 atm)	(lb/ft^3)	49.60	49.30	52 to 55
	(lb/gal)	6.630	6.590	7.0 to 7.3
Boiling point (°F @ 1 atm)		148.2	172.9	325 to 750
Freezing point (°F)		-143.8	-173.4	< 20
Vapor pressure (psi @ 100°F)		4.631	2.313	negligible
Heat of vaporization (Btu/lb)		462.6	361.4	116
Gross heating value (Btu/lb)		9,760	12,800	19,300
Net heating value	(Btu/lb)	8,600	11,600	18,100
	(Btu/gal)	57,000	76,200	128,000
Stoichiometric mixture net heating value (Btu/lb)		1,152	1,157	—
Autoignition temperature (°F)		867	685	494
Adiabatic flame temperature (°F)		3,400	3,480	3,620
Flame speed @ stoichiometry (ft/s)		1.4	—	—
Octane –Research		106	107	—
–Motor		92	89	—
Cetane		0 to 5	0 to 5	> 40
Flash point (°F)		52	55	126 to 204
Flammability limits (vol% in air)		6.72 to 36.5	3.28 to 18.95	1.0 to 5.0
Stoichiometric air/fuel mass ratio		6.46	8.98	14.5
Stoichiometric air/fuel volumetric ratio		7.15	14.29	85
Sulfur content (wt%)		0	0	< 0.05

alcohols produces formaldehyde and acetaldehyde, compounds that are both toxic and highly reactive in forming ozone. The bulk of aldehyde emissions with methanol engines are formaldehyde, while ethanol produces more acetaldehyde. Alcohols' resistance to autoignition have traditionally made them suited for spark-ignition engines. Spark-ignition engines can operate more efficiently by taking advantage of alcohols' high octane through increased compression ratios. Compression-ignition engines running on alcohol usually require modifications that optimize the autoignition of alcohols. Compression ratios typically are raised above comparable diesel levels, air induction modified, and ignition aids such as spark plugs, glow plugs, and ignition improvers used for starting or low load operation.

Of the two alcohols considered here, methanol and ethanol, more heavy-duty engine development work has been performed on methanol. Methanol has a lower energy density than diesel fuel; 2.25 gallons of methanol has about the same energy as one gallon of diesel fuel. Methanol is corrosive and has a lower lubricity than diesel fuel, requiring appropriate methanol-compatible materials to be used in the fuel system. Methanol reacts with most standard diesel engine oils, requiring the use of a low-ash oil to maintain engine lubrication plus an additive package designed for methanol.

Ethanol can also be used as an engine fuel. It also has a lower energy density than diesel fuel, through not as low as methanol's: 1.68 gallons of ethanol has about the same energy as a gallon of diesel fuel. Ethanol is not as corrosive as methanol but still has a lower lubricity than diesel fuel, requiring that the fuel system be made of ethanol-compatible materials.

Three common variations of methanol and ethanol are M100, M85 and E95. M100 is neat methanol with no gasoline added. M85 is 85 percent neat methanol and 15 percent gasoline. E95 is 95 percent ethanol and 5 percent gasoline. Gasoline has been added to methanol and ethanol to enhance cold starting in light-duty vehicles and flame luminosity of the fuels.

In order to operate a diesel engine on an alcohol fuel, the engine must be designed for the properties of the new fuel. Some of these changes have already been noted above, such as using alcohol-compatible materials. The fuel system must also be modified to increase the volume of fuel injected to make up for the lower energy density of alcohols. Some means of igniting the charge is required because of the low cetane number of alcohols. Higher compression ratios have been used to reduce ignition delay and improve emissions.

Methanol is toxic if ingested. Ingestion of 6 ml can cause visual disturbance, pain, nausea and vomiting. Ingestion of 100 to 250 ml of methanol can cause death. Methanol in its pure form does not have a strong odor or taste, but in the case of M85, it does have a strong odor and taste due to the presence of gasoline. This helps defer accidental ingestion. Odor and taste additives can be added to M100 as a safety precaution. Ethanol is much less toxic than methanol, but can cause death if enough ethanol is ingested. Nausea and vomiting can also result from ingestion of smaller quantities.

Flame luminosity of alcohol fuels is also considered a safety risk. Alcohols burn very cleanly, thus alcohol flames are difficult to see under bright sunlight. Gasoline or other additives make the flame more visible.

The following subsections describe different engine technologies which can use alcohol fuels. The following technologies are covered: two-stroke, direct-injection engines; four-stroke, direct-injection, ignition-assisted engines; ignition improvers; and four-stroke, spark-ignited Otto-cycle engines. Each of these technologies is considered in terms of its emissions, fuel economy, durability and reliability, and technical feasibility.

To make hydrocarbon emission results consistent with diesel hydrocarbon emissions, an organic matter hydrocarbon equivalent (OMHCE) emission standard was developed by EPA. OMHCE calculates the weight of oxygenated hydrocarbon and aldehyde emissions without the weight of the bound oxygen atom. Emission results for heavy-duty methanol engines are generally reported as OMHCE.

The ARB has adopted formaldehyde standards for heavy-duty methanol engines. For 1993, formaldehyde is limited to 0.1 g/bhp-hr. In 1996, the standard is reduced to 0.05 g/bhp-hr. Use of oxidation catalysts with methanol engines should reduce formaldehyde emissions below these standards.

5.1 TWO-STROKE, DIRECT-INJECTION ENGINES

Two-stroke engines have shown much promise as alcohol heavy-duty engines. Two-stroke engines offer higher output compared with a four-stroke engine of the same size because they have a power stroke every crankshaft revolution instead of every other revolution. Valving is also simplified somewhat because two-stroke engines have intake ports on the cylinder wall rather than intake valves in the head. Another aspect different from four-stroke engines is that, following the opening of an exhaust valve or port, two-stroke engines clear the combustion products from the cylinder through scavenging. Scavenging is accomplished in these engines by forcing the exhaust from the engine cylinder with boosted intake air (Reference 29).

Detroit Diesel Corporation (DDC), which produces 70 to 80 percent of the engines used in urban transit buses, has developed methanol versions of their two-stroke engines. The DDC 6V-92TA engine is the most common transit bus engine and has had the most development work on designing it for methanol. The methanol version of this engine is now certified to meet California's 1991 Bus standards. The 6L-71 and the 149 are used in a variety of industrial and military applications. The 6L-71 methanol engine is currently being demonstrated in a trucking application and 149 is being demonstrated in an electrical generating set. All these engines are direct-injected, blower-scavenged, turbocharged, and aftercooled. Both the 6V-92 and the 6L-71 utilize a high compression ratio and glow plug assistance during startup. Autoignition after the engine is warmed up is achieved by the high compression ratio and reduced scavenging. Both increase the in-cylinder temperature to autoignite the fuel charge. Reduced scavenging also has the beneficial effect of reducing NO_x emissions much like EGR.

5.1.1 DDC 6V-92TA Engine

In the summer of 1991, the DDC 6V-92TA became the first heavy-duty methanol engine to receive certification by the ARB and the EPA. The 6V-92TA, a "vee" configuration with six 92 cubic inch displacement cylinders, is available in 253 bhp or 277 bhp coach engines, and is currently undergoing field testing in the 300-350 bhp range suitable for heavy-duty trucks. As mentioned above, the engine is turbocharged, aftercooled, and blower-scavenged. It uses a compression ratio of 23:1 and glow plugs to facilitate ignition during starting. The glow plugs are used if the coolant temperature is less than 160°F, and for one minute after a restart, regardless of coolant temperature. Thus glow plugs are used seldom except during warmup and occasionally during idle or after a restart. The electronic unit injectors and blower bypass are electronically controlled by the Detroit Diesel Electronic Control (DDEC II) system which controls injection timing and duration. The 6V-92TA makes use of reduced scavenging (as compared to the diesel counterpart) to help counter the high heat of vaporization of alcohol fuels (Reference 30). In addition, the intake port height, exhaust valve cam profile, and injector tips have been optimized for methanol use (Reference 31). To provide additional emissions control, the production version of the methanol 6V-92TA includes a platinum oxidation catalyst with a ceramic monolith measuring 9 inches in diameter, and 6 inches in length (Reference 32). The catalyst is mounted adjacent to and downstream of the turbocharger. This catalyst significantly reduces engine-out unburned methanol and formaldehyde emissions.

The 6V-92TA was certified in 1991 on M100 at two power levels and also on M85. The engine configurations are as noted above. The results of the tests, with deterioration factors, are shown in Table 5-2. Two of these configurations almost meet Scenario 1 levels. The CO emissions on these engines, while within the 15.5 g/bhp-hr standard, can be lower with this technology. As discussed later, DDC is working to improve CO and NO_x emissions from this engine.

Table 5-3 shows emission results from the same engine operating on M100 without a catalyst and on E95 with a catalyst. DDC has also tested versions of the 6V-92TA with power ratings suitable for heavy-duty truck engines, in the 300 to 350 hp range. These engines were tested over the FTP hot cycle transient test using a 21:1 compression ratio. The engines were tested on M100

Table 5-2. 1991 certification results for 6V-92TA coach engine with catalyst (g/bhp-hr)[a]

Fuel	Power (hp)	OMHCE	CO	NO_x	PM	Aldehydes
M100	253	0.53	7.75	2.26	0.050	0.061
M85	253	0.76	11.97	3.03	0.056	0.061
M100	277	0.41	4.75	2.35	0.057	nm[b]

[a]Includes deterioration factors
[b]nm = not measured

Table 5-3. 1991 emission results for 6V-92TA coach engine (g/bhp-hr)[a]

Fuel	Power (hp)	OMHCE	CO	NO_x	PM
E95[b]	253	1.80	23.07	3.74	0.162
M100[c]	277	1.39	7.62	3.91	0.105

[a]Includes deterioration factors
[b]With catalyst
[c]Without catalyst

and M85, with and without catalysts (particulates were not measured on the tests with catalysts). The results are shown in Table 5-4 (Reference 33).

DDC methanol 6V-92 bus engines have accumulated over 4,500,000 miles of field experience with 53 of the 63 field units in the production configuration. Fuel injector life increased dramatically with the addition of the Lubrizol fuel additive, which alleviated tip plugging. Increasing the compression ratio to 23:1 lengthened glow plug life by increasing in-cylinder temperatures and reducing glow plug activation time. On the average, glow plugs perform without incident for 60,000 miles. Catalytic converter life is an issue: converter failures have occurred in the field, typically in conjunction with glow plug malfunction or injector damage. The primary cause of the failures is rapid temperature fluctuations within the catalyst which produce thermal stresses and can shatter

Table 5-4. FTP hot cycle emission test results for 6V-92TA truck engine configuration (g/bhp-hr)

Fuel	Catalyst	Power (hp)	NO_x	CO	PM	Aldehydes	OMHCE
M100	No	350	2.6	3.8	0.20	0.14	2.0
M100	No	300	2.9	3.2	0.11	0.12	2.2
M100	Yes	300	2.6	0.3	nm[a]	0.06	0.2
M85	No	300	3.4	6.7	0.09	0.12	2.2
M85	Yes	300	3.4	1.1	nm	0.07	0.06

[a]nm = not measured

the ceramic monolith. In the absence of other failures, however, durability testing shows converters lasting over 100,000 miles. DDC is developing diagnostics which will shut down the engine if the catalyst overheats. DDC is also participating in a number of methanol truck demonstrations with six trucks in California and one in New York City (Reference 34). These trucks entered service in 1990.

Two of the largest DDC methanol bus demonstration sites are in Los Angeles, California with the Southern California Rapid Transit District (SCRTD) and in New York City with the NYC Department of Transportation (NYCDOT). In the fourth quarter of 1991, the SCRTD methanol fleet average fuel economy was 1.13 mpg (2.54 mpg diesel equivalent) compared to the diesel fleet average fuel economy of 3.09 mpg (Reference 35). The NYCDOT methanol fleet average fuel economy for the first quarter of 1991 was 1.24 mpg (2.8 mpg diesel equivalent) compared to the diesel fleet average fuel economy of 3.2 mpg (Reference 36). Another California demonstration, at the Riverside Transit Agency, reported a 1991 average fuel economy of 1.47 mpg (3.3 mpg diesel equivalent) for their methanol fleet of three buses compared to the average diesel fleet fuel economy of 3.4 mpg (Reference 37). Riverside Transit methanol buses are used in service that requires less stops and higher average speeds than either SCRTD or NYCDOT buses, which are used in highly urban service, and thus show improved fuel economy in both methanol and diesel versions.

The brake thermal efficiency of the production configuration methanol 6V-92TA compares more closely with the diesel engine under steady-state rated operating conditions. The methanol version is slightly lower (one to three percent) than the diesel over a wide range of engine speeds. DDC attributes this lower efficiency to the low scavenging ratio necessary to meet the 1991 HC standards without exhaust aftertreatment. With several development engines, DDC has demonstrated a methanol-fueled brake thermal efficiency equivalent to that of the counterpart diesel-fueled engine and believes that adequate HC control might be achieved in the near future without penalizing the fuel efficiency of the methanol engines (Reference 38).

DDC continues to refine the 6V-92TA to meet tighter standards. Their 1993 certification results are shown in Table 5-5 for M100, M85 and E95. DDC has controlled particulate emissions through better engine oil control, and the NO_x through a retarded injection timing strategy. NO_x for the M100 engine are well below Scenario 2 levels while the emissions for M85 and E95 increased in this configuration. As this engine was optimized for M100, additional engine tuning could be performed to reduce NO_x emissions on the other fuels. Particulate emissions in the methanol engine consist of 80 to 90 percent engine oil, compared with an estimated 30 to 40 percent engine oil contribution in diesel engines. The retarded injection timing strategy, while reducing NO_x, increased HC and fuel consumption. HC and formaldehyde emissions were

Table 5-5. 1993 certification results for 6V-92TA coach engine with catalyst (g/bhp-hr)

Fuel	HC	CO	NO_x	PM	Aldehydes
M100	0.10	2.00	1.70	0.030	0.07
M85	0.20	1.60	4.10	0.030	0.08
E95	0.70	1.70	4.20	0.040	0.02

controlled by the catalyst, however, fuel consumption increased 4 percent to 6 percent over the 1991 version. Improved injection timing strategies, better air system matching, and ceramic combustion chamber components for improved heat retention in the cylinder could further improve efficiency.

5.1.2 DDC 6L-71TA Engine

The 6L-71TA is an in-line 6 cylinder engine with 71 cubic inch displacement in each cylinder. Its hardware configuration is similar to that of the 6V-92TA, listed above. The 6L-71TA was tested over the FTP using a 23:1 compression ratio and a modified blower to reduce scavenging. Table 5-6 shows the results.

There is limited experience with the 6L-71TA in demonstration use. One engine is being demonstrated in a Federal Express tractor in Los Angeles. This engine has averaged a nine percent lower diesel equivalent fuel economy than its diesel counterpart which was powered by a four-stroke engine.

The 6L-71TA is already well under the Scenario 1 and 2 goals for particulates. More work in injection timing optimization is required to reduce the NO_x levels further. DDC development efforts will benefit from those on the 6V-92TA.

5.1.3 Two-Stroke, Direct-Injected Engine Summary

DDC has reached Scenario 1 goals with its M100 version of the 6V-92TA coach engine. This engine is currently available in the 253 and 277 horsepower versions. DDC has stated that they are considering also offering a 300 horsepower version. While these engines are suited for

Table 5-6. FTP hot cycle emission test results of 6L-71TA 300 hp truck engine configuration on M100 (g/bhp-hr)

Catalyst	CO	NO_x	PM	OMHCE	Aldehydes
No	3.5	3.3	0.12	1.8	0.16
Yes	0.5	2.7	0.02	0.1	0.05

transit buses and medium heavy-duty trucks, larger engines are needed for the heavy heavy-duty class. The DDC 8V-92TA engine could be developed for higher power applications.

5.2 FOUR-STROKE, DIRECT-INJECTION, IGNITION-ASSISTED ENGINES

The majority of heavy-duty diesel engines are four-stroke direct-injection engines. Subsequently, several of the major methanol engine developments have focused on modifying existing four-stroke diesel engines and adapting them to use methanol. These engines have been redesigned to take into account the various attributes of methanol already discussed, among which, the chief concern is how to compensate for methanol's long ignition delays due to its low ignition quality while benefiting from its clean burning properties.

5.2.1 Four-Stroke, Direct-Injection, Ignition-Assisted Engine Technology

Development of heavy-duty methanol engines has followed two approaches. The first uses hot surface ignition generated by glow plugs. The second utilizes spark plugs to ignite the fuel/air mixture. Regardless of the approach taken, the engines share and make use of similar concepts which are described below.

Due to methanol's 2.25:1 diesel equivalent energy density, fuel injection systems are sized to increase fuel capacity to maintain engine performance. Fuel pump sizes and injector plungers, larger than their diesel counterparts, are used, while adjustments to plunger stroke, plunger pre-stroke, and injector pressure are optimized to allow adequate fuel delivery. Problems have arisen related to the gumming of injectors and injector pumps because of methanol corrosion and inadequate methanol compatible lubrication. To overcome these problems, special engine lubricants and lubrication systems are used. Other engine components, likewise susceptible to methanol corrosion, are replaced with parts fabricated from more resistant materials.

Glow plugs, for hot surface ignition, and spark plugs are utilized to provide ignition sources. Electronic control units (ECU) currently modulate the energy supplied to glow plugs based on engine speed, engine load, coolant temperature, and ambient temperature to maximize efficiency and fuel economy. Specifically, at low loads, combustion temperatures are low and more energy is required for ignition. Conversely, under heavy loads, the combustion temperatures are high and reduce the need for external energy. By careful monitoring of glow plug temperatures, glow plug life can be maximized and HC and CO emissions limited.

Caterpillar, Deutz, and Navistar use glow plugs in their methanol engines. Early glow plug technology was susceptible to high rates of failure due to thermal fatigue which led to cracking of the glow plug. Wide temperature fluctuations are created due to methanol cooling effects as the fuel mixture was injected into the piston bowl very near the glow plugs. The methanol spray cools the glow plugs which are then immediately subjected to combustion heat, incurring severe thermal stresses. To alleviate these stresses, ceramic glow plugs with perforated shields have been designed. The shield isolates the plug from high combustion temperatures and prevents sudden temperature drops of the surface due to fuel contact. These shielded glow plugs are being used in Navistar and Deutz engines. The glow plug shield also improves fuel vaporization and combustion.

Injection timing control is being used to overcome the problems created by poor combustion due to long ignition delays. At light loads, the start of injection is advanced to lower CO and HC emissions. However, at medium and heavy loads, the injection timing is retarded for low NO_x emissions. In the D2566 methanol engine, M.A.N. replaced the mechanical distributor with an electronic distributor but used a fixed timing of 8 crank angle degrees before top dead center (BTDC). Emissions could be further optimized with variable timing. Other manufacturers have developed successful injection timing though ECUs which permit timing variations based on instantaneous engine demands.

Higher compression ratios are used to reduce ignition delay and increase charge temperatures at the end of compression. These factors decrease HC and CO, especially at low loads, while increasing NO_x and fuel economy. Compression ratios over 18:1 have been used versus a typical 16:1 ratio for diesel engines.

Another means to overcome methanol's ignition resistance is to preheat the intake air. Preheating can be accomplished with an air/coolant intercooler. This effectively raises the fuel mixture charge temperature which improves ignition characteristics as well as accelerates combustion after ignition has occurred.

High PM emissions have been attributed to poor oil control within these engines. To address this problem, new piston rings and the use of valve stem seals have been implemented. Further redesign and optimization of these parts should further reduce PM emissions in the future.

Platinum oxidation catalysts are being employed as aftertreatment to reduce HC, formaldehyde and CO emissions. Research is being performed to optimize aftertreatment processes to work efficiently at methanol engine exhaust temperatures. Another concern is catalyst failures. When improper combustion occurs, unburnt fuel enters the catalyst where it can reduce efficiency or even destroy the catalyst through overheating. This catastrophic failure of a catalyst can result in high exhaust back pressures which can lead to engine damage. To avoid this problem, catalyst sensors are employed to signal when the convertors are not operating properly.

Exhaust gas recirculation (EGR) is under development to aid in the reduction of NO_x, and HC under partial loads. The general practice is not to use EGR unless an engine is turbocharged because turbocharging keeps the mixture from becoming too rich. This keeps NO_x production low by diluting the charge with EGR and HC emission low through fuel leaning. Since methanol burns virtually soot free, EGR technology can be applied over a wide band of engine operating conditions. Research by Navistar, FEV, Hino and others using EGR rates of up to 60 percent has shown significant reduction of NO_x.

5.2.2 Four-Stroke, Direct-Injection, Ignition-Assisted Engine Emissions

The results of various four-stroke, direct-injection engines are presented below in Table 5-7 (Reference 39, 40, 41, 42). None of the engines made use of EGR which would have further reduced NO_x emissions. As evident by the HC and CO emissions, oxidation catalysts are important in future applications if emission standards are to be met. The benefits of an ECU controlling injection timing, glow plug modulations, and EGR would allow for a dynamic systems optimization resulting in better emissions and performance characteristics.

Table 5-7. Emissions from four-stroke, direct injection, ignition-assisted engines on M100 (g/bhp-hr)

Engine	Ignition	Oxidation Catalyst	Emission Test	PM	NO_x	CO	OMHCE	Aldehydes
M.A.N. D2566-UH	Spark plugs	Yes	13 mode	—	5.75	0.28	0.16	—
Caterpillar 3406B	Glow plugs	No	13 mode	—	2.65	4.71	2.38	0.07
Caterpillar 3406B	Glow plugs	No	FTP	0.15	3.05	12.40	4.45	0.49
Navistar DT466	Glow plugs	No	FTP	0.12	4.70	3.40	1.06	—
Navistar DT466	Glow plugs	Yes	FTP	0.05	3.52	0.06	0.06	—

The emission results indicate that the 1994 PM and NO_x standards of 0.1 and 5 g/bhp-hr are almost met with these development engines. In fact, the Navistar DT466 utilizing an oxidation catalyst falls within the acceptable standards. However, none of these engines with their current test configurations were able to approach the Scenario 1 goals of 0.05 and 2 g/bhp-hr for PM and NO_x.

In order to meet these desired standards, methanol engine development will have to continue in the areas discussed above. It seems reasonable that with optimized injection timing, application of EGR, refinements in the ECU and ignition systems that the technology will be able to meet Scenario 1 goals. With methanol's inherently low PM emissions, the major technical challenge is NO_x reduction. The most promising strategy with methanol engines is to use EGR and retarded injection timing to control NO_x and reduce HC, formaldehyde and CO emissions using an oxidation catalyst.

5.2.3 Four-Stroke, Direct-Injection, Ignition-Assisted Engine Fuel Economy

Overcoming methanol's resistance to ignition requires that an external energy source be utilized during engine operation. Moreover, to promote proper combustion during idling, engine RPM is typically set about 20 percent higher than for standard diesel engines. These additional energy demands decrease vehicle fuel economy.

To retain comparable driving range between refueling, methanol powered vehicles require larger fuel tanks which in turn increase the weight of the vehicle. This additional weight is on the order of 1.5 percent of vehicle weight which can also affect fuel economy when compared with diesel-powered vehicles.

As a result of the additional energy requirements, higher idling settings, and the increase in vehicle weight, methanol-fueled vehicles currently have approximately 5 percent worse diesel equivalent fuel economy. It is important to realize that these converted engines have not been optimized to make use of methanol's fuel characteristics. Instead, research and development focused on minimal engine modification to reach diesel performance levels while reducing emissions. The increased use of methanol as a vehicle fuel will promote future OEM research to optimize engines with respect to emissions, fuel consumption, and performance.

5.2.4 Four-Stroke, Direct-Injection, Ignition-Assisted Engine Durability and Reliability

While four-stroke methanol engines have proven themselves in field demonstrations, the reliability of some components need improvement for commercial operation. Glow plugs initially suffered similar shortcomings, but new ECUs have been able to better regulate and reduce glow plug wear. The advent of protective shielding has also significantly increased reliability. The average life of spark plugs is in the range of 5,000 to 10,000 miles, which is still significantly below reasonable spark plug life. This results from the need to generate more ignition energy with methanol fuels as well as from wear due to the harsher environment of higher temperatures and pressures than typical spark-ignited engines.

Piston crown cracking initially resulted from wide cyclic temperature ranges due to methanol cooling at the point of injection. Pistons have been redesigned to minimize these effects. Caterpillar, for example, incorporated a central steel impingement pin to relieve the crown of thermal shock effects from cooling, as well as aiding in fuel spray breakup.

Ultimately, none of these issues have proven unsurmountable by current technology. One example of this is a Peterbilt 320 refuse hauler which has been outfitted with a Caterpillar 3306

DITA. As of February, 1992 this vehicle had logged over 21,000 miles with an average uptime of 92 percent. Initially, the engine suffered from glow plug failures, but these were eventually stopped with a new voltage control unit that reduced voltage spikes and unnecessary wear.

5.3 IGNITION IMPROVERS

Alcohols have very low cetane numbers, making them difficult to ignite in compression-ignition engines without modifications. One means of ignition is to use cetane enhancers or ignition improvers, i.e. chemicals that are mixed with the alcohol to create a fuel which will ignite on compression. Alcohols, with an ignition improver, may be used in diesel engines without the electrical modifications found in other methanol engines. The rate of fuel injection must be increased, like in the spark or glow plug assisted engines, to achieve high enough torque and power with alcohol's low energy density, and the fuel system must also be modified with alcohol-compatible materials. Because ignition improvers work as "chemical spark plugs," they are well suited for diesel engine retrofits. By using ignition improvers, the advantages of alcohols can be obtained without adding an electrical ignition system.

Currently there are two different types of ignition improvers: those that are fumigated into the cylinder as a gas and those that are mixed with the fuel in the tank. An example of the first type is dimethyl ether (DME) (although DME can also be mixed with the fuel). Examples of the second type are Avocet, 2-ethylhexyl nitrate, and tri-ethylene glycol dinitrate.

5.3.1 Dimethyl Ether

DME can be produced on-board the vehicle by the dehydration of methanol, using an exhaust-heated catalyst, and then fumigated into the air intake prior to the intake port. A high proportion of DME (over 50 percent of the fuel mixture) is required to help the fuel mixture ignite at idle in an unmodified engine. As engine speed and load increase, the proportion required decreases. At low loads, DME improves the consistency of ignition and reduces the ignition delay compared to glow plugs.

Because DME is fumigated into the intake air charge, it is present in the cylinder before the injection of the fuel. Recent research has shown that the DME starts to combust before the point of injection and is mostly burned off before TDC, resulting in negative work and increased NO_x (Reference 43). Further, the combustion of DME before the injection point reduces the burning rate of the injected fuel because less oxygen exists in the cylinder. This contributes to a lower thermal efficiency. Because of these factors, the use of DME as a fumigated ignition improver will require further development work. However, recently DME has gained popularity as a fuel additive.

5.3.2 Avocet

Avocet is an additive package for alcohol fuels produced by ICI that includes an ignition improver, a lubricity additive, and a corrosion inhibitor. Its chemical properties are shown in Table 5-8. Avocet is a nitrate-ester formulation blended with methanol and can be mixed with alcohols in concentrations of one to five percent to produce a fuel mixture that will ignite in a compression-ignition engine without using spark plugs or glow plugs. In addition to the engine modifications required for the use of an alcohol with an ignition improver noted above, further

Table 5-8. Properties of Avocet

Property	Value
Molecular weight	32 to 300
Specific gravity	1.15
Density (lb/gal)	9.6
Boiling point (°F)	149
Vapor pressure (psi @ 100°F)	4
Net heating value (Btu/lb)	8,040
(Btu/gal)	77,000
Autoignition temperature (°F)	<400
Flash point (°F)	50

modifications can improve combustion and emissions. Increasing the compression ratio increases cylinder temperature and pressure, and thereby decreases the concentration of Avocet required in the fuel. Advancing the injection timing also improves the idle and cold starting.

DDC tested alternative-fuel versions of the 6V-92TA using both methanol and ethanol with Avocet (Reference 44). The engine tested was a pre-certification version of the 6V-92TA. This DDEC controlled engine had a compression ratio of 23:1, a shorter exhaust camshaft duration, a shorter liner port height to increase the effective compression ratio and reduce the engine air flow, and a blower with a large electronically-controlled bypass valve. In addition, the engine had a turbocharger sized to the reduced air flow, and the cylinder head was changed to allow glow plugs for improved starting and light-load performance. Finally, the electronic unit injectors had modified tips to accommodate the higher flow of alcohol fuel required to produce the same power as the diesel version of the engine.

Both methanol and ethanol with one percent Avocet were tested over the FTP. Table 5-9 shows the emissions results. The engine was optimized for operation on M100, and the only change to the engine configuration during these tests was that the glow plugs were turned off when the engine was operating with Avocet. These results show that the use of Avocet increased emissions of NO_x by 39 percent and particulates by 20 percent, while reducing the emissions of OMHCE by 34 percent, CO by 29 percent, and aldehydes by 50 percent. There was a slight change in thermal efficiency. The increases in NO_x and PM emissions and decreases in HC and CO might be attributed to reduced ignition delay and higher combustion temperatures at high loads from the use of Avocet throughout the load map of an engine optimized for the use of ignition assistance only at low loads. Using ethanol with Avocet produced a 9 percent reduction in NO_x, a 12 percent reduction in OMHCE, no change in particulates, a 13 percent increase in aldehydes, and a 144 percent increase in CO. Thermal efficiency was reduced 6 percent. As these results were

Table 5-9. FTP transient emissions of a 6V-92TA engine on methanol and ethanol with 1 percent Avocet (g/bhp-hr)

Fuel	CO	NO_x	PM	OMHCE	Aldehydes	Peak Thermal Efficiency (%)
M100	2.42	1.94	0.10	1.95	0.150	35.8
MeOH/A	1.72	2.69	0.12	1.28	0.075	36.0
EtOH/A	5.91	1.77	0.10	1.71	0.170	33.6

MeOH/A = 99 percent methanol + 1 percent Avocet
EtOH/A = 79.2 percent ethanol + 19.8 water + 1 percent Avocet

obtained for an engine that was optimized to operate on M100, further improvements in emissions could be obtained if the engine was calibrated for operation on the specific fuel used.

Avocet has been used as a cetane improver in many alcohol vehicle demonstrations throughout the world. An ongoing demonstration using transit buses is the Methanol and Avocet Supporting Southern California Air Resources (MASSCAR) Project at the Southern California Rapid Transit District (SCRTD) (Reference 45). The fleet consists of 12 buses that have accumulated over 250,000 miles operating on methanol and Avocet (Reference 46). These buses use retrofitted DDC 6V-92TA engines, modified with an increased compression ratio, high flow mechanical fuel injectors, and a lower capacity blower. The higher compression ratio of 23:1 (compared with 17:1 for the standard diesel turbocharged version) increases temperatures at the beginning of injection; therefore less Avocet is required for stable combustion. Also, the injection timing was retarded to smooth idle operation, enhance cold starting and lower emissions. Because the lower capacity blower reduces scavenging in the cylinder, there is a larger amount of combustion residuals left in the cylinder, further increasing the cylinder temperature at the start of injection. The blower on the 6V-92TA has a bypass valve which opens at high turbocharger boost pressures to reduce scavenging at high loads. The bypass valve on the methanol/Avocet version is modified to open at lower boost pressures, further reducing scavenging at intermediate loads. An oxidation catalyst is used to reduce HC, formaldehyde and CO emissions. Recent test results from this engine operating on methanol and Avocet are given in Table 5-10 (Reference 47).

Most recent fuel economy data shows that the methanol/Avocet buses are achieving an average fuel economy of 1.3 mpg (2.92 mpg diesel equivalent) compared to the diesel control buses average fuel economy of 2.83 mpg. The buses average about 2,500 miles per month, compared with approximately 3,500 miles per month by the diesel versions. The methanol buses have about the same number of road calls as the diesel control buses, convincing SCRTD of their reliability. SCRTD started to integrate the methanol/Avocet bus maintenance into the regular bus fleet maintenance during the third quarter of 1991.

Another demonstration of methanol/Avocet was in an asphalt truck run by the City of Los Angeles. This truck was part of the California Methanol Fueled Truck Demonstration sponsored by the California Energy Commission and the South Coast Air Quality Management District. The truck has a Cummins L10 engine which is a four-stroke, direct-injected, turbocharged, aftercooled, in-line six-cylinder engine with an oxidizing catalytic converter. The methanol version of the L10 has been under development by Cummins since 1985 and is very similar to the diesel version. The only major change is that the fuel system has been modified to be methanol compatible and to give a higher rate of flow necessary to produce the same torque and power as the diesel engine. This engine also has ceramic-coated pistons to stop cracking of the piston crown found in earlier demonstrations.

Table 5-10. FTP emissions of a 6V-92TA engine on methanol with 1 percent Avocet (g/bhp-hr)

CO	NO_x	PM	OMHCE
0.61	3.96	0.039	0.22

The vehicle operated for over two years and accumulated 18,976 miles and 1,119 engine hours. Throughout the demonstration, Cummins made various improvements to the fuel system. These improvements have lengthened injector life from about 2,500 miles to about 4,000 miles. The vehicle's fuel economy was about one percent higher (diesel equivalent) than that of the diesel control vehicle in their most recent service route. The truck was run on the ARB eight-mode steady-state chassis test in December of 1991. The results are shown in Table 5-11.

The methanol version of this engine will probably not receive further development. This vehicle experienced troubles with idle fuel control which was due to fuel vaporization in the injection pump, and resulted in a large amount of downtime. Cummins has set the development of the natural gas version of the L10 as a higher priority.

There have been other demonstrations of methanol and Avocet in North America, Europe and New Zealand that have shown this configuration's technical feasibility, relatively low emissions, good fuel economy and durability.

5.3.3 Ignition Improver Summary

Ignition improvers offer an alternative means to overcome methanol's resistance to ignition, giving manufacturers or users an option other than electronic ignition methods. Ignition improvers are especially important for retrofitting buses to operate on alcohols such as methanol because they negate the need to add an electrical system to operate glow or spark plugs. Emission tests have shown that the benefits of methanol use can be gained through a methanol and Avocet mixture; however, not much development work has been performed to reduce these emissions below the

Table 5-11. Steady-state 8-Mode emission results with a Cummins L10 operating on methanol with 5 percent Avocet (g/bhp-hr)

CO	NO_x	PM	HC
0.063	6.01	0.018	0.026

1994 heavy-duty diesel emissions. Judging from the DDC work on the 6V-92TA, engines running on methanol with ignition improvers should be able to make the Scenario 1 goals, providing they have been modified to include all of the techniques that help lower emissions, such as turbocharger and blower control, high compression ratios, retarded injection timing, and catalysts.

5.4 FOUR-STROKE HOMOGENEOUS CHARGE ENGINES

Another method of using alcohol fuels in diesel engines is to convert the engine to a standard Otto-cycle engine. This requires that the fuel be introduced into the intake manifold and that the direct-injection fuel injectors be replaced with spark plugs and an ignition system. An example of such a conversion is discussed below.

5.4.1 Four-Stroke Homogeneous Charge Engine Technology

In 1988, Ford converted a Ford-New Holland 6.6L turbocharged, compression-ignited, direct-injection, four-stroke diesel engine to operate on methanol. The methanol 6.6L differs from the diesel significantly; the methanol engine is naturally-aspirated, spark-ignited, and port-injected. The Ford 6.6L is the only heavy-duty methanol engine to date which utilizes port-injection (homogeneous charge). The compression ratio of the methanol engine was reduced from 16.7 to 11.3 to prevent preignition. Additionally, the cylinder head was machined for spark plug installation, the intake manifold was redesigned to provide mounting for the electronic sequential port fuel injectors, and modifications were made to the Electronic Engine Control (EEC IV) system. Finally, the camshaft was redesigned to achieve the power and torque characteristics similar to the diesel engine. The Ford methanol engine was field tested in Los Angeles in an Arrowhead water delivery truck.

5.4.2 Four-Stroke Homogeneous Charge Engine Emissions

Southwest Research Institute (SwRI) performed EPA transient emission testing for Ford during the summer of 1990. The engine tested was identical to the engine being demonstrated in the Arrowhead delivery truck. The test fuel was M85.

Ford chose to calibrate the 6.6L methanol engine to meet gasoline-fueled heavy-duty Otto-cycle engine emission standards because the 6.6L methanol engine is naturally aspirated and spark-ignited. Thus, the procedures and test schedules followed were those prescribed for gasoline engines rather than those for diesel engines.

Emissions results are shown in Table 5-12 (Reference 48) in comparison to 1991 heavy-duty gasoline and diesel engine standards. All results are converter-out emissions; the engine was equipped with a three-way catalyst.

The Ford 6.6L engine utilizes a three-way catalyst system and operates near the chemically correct mixture for complete combustion (stoichiometric air/fuel ratio). This has a dramatic effect on all emissions, particularly NO_x, HC and aldehyde emissions. The CO emissions are relatively high and most likely the result of rich operation at full power conditions to maintain the torque and power of the diesel engine it replaces.

5.4.3 Four-Stroke Homogeneous Charge Engine Durability and Reliability

The low compression ratio, homogeneous charge, and spark-ignition combination employed in the Ford methanol engine potentially has a positive effect on engine durability. This combination results in lower in-cylinder peak temperatures and pressures, reducing thermal and pressure stresses on the engine components. Ford conducted a 300 hour durability test under full load and experienced no engine problems. The 6.6L diesel engine is a 2000-hour engine and Ford expects a similar lifetime for the methanol engine.

Table 5-12. Ford 6.6L methanol engine test results with three-way catalyst (g/bhp-hr)

	PM	NO_x	HC	CO	Aldehyde
1991 Diesel Standards	0.25	5.00	1.30	15.5	—
1991 Gasoline Standards	—	5.00	1.90	37.1	—
Ford 6.6L Engine	—	1.21	0.32	21.1	0.011

The methanol 6.6L performed well in the field; the total in-service time to date is 1,141 engine hours, and uptime since the vehicle entered service was approximately 90 percent. The only engine difficulty experienced in the field was a spark plug failure at 12,718 miles. The spark plugs were replaced and no further problems occurred.

5.4.4 Four-Stroke Homogeneous Charge Engine Fuel Economy

The throttled homogeneous charge approach has a lower efficiency than direct-injected diesel engines, but offers high power output. The Arrowhead demonstration vehicle has averaged approximately 6.1 miles per diesel equivalent gallon over the course of the demonstration. This is about 9 percent worse than the fuel economy exhibited by the counterpart diesel engine.

5.4.5 Four-Stroke Homogeneous Charge Engine Summary

The Ford methanol 6.6L engine falls well within the Scenario 1 NO_x goals. With further development, the CO emission could be reduced to meet diesel standards with some impact on NO_x. As the gasoline test procedure does not require particulate measurement, no information on the particulate emissions is available, however, experience with four-stroke methanol engines suggest levels below 0.05 g/bhp-hr. The low compression ratio strategy in conjunction with a stoichiometric/TWC appears to reduce NO_x to low levels without incurring large losses is fuel economy.

5.5 ALCOHOL FUEL TECHNOLOGY SUMMARY

The FTP emissions from the alcohol engines discussed in this section are shown in Table 5-13 and plotted in Figure 5-1. Four engines have NO_x levels at or below 2 g/bhp-hr. Methanol engines show promise for meeting the Scenario 1 goals. DDC has developed an engine under 2 g/bhp-hr NO_x and 0.05 g/bhp-hr PM and is offering it for sale in California for transit buses. The four stroke engines show some promise but will require more work to achieve such low emissions. Ignition improvers are another option to help ignite methanol, but so far, not much development has been done on new engines relying solely on ignition improvers. Finally,

Table 5-13. Survey of heavy-duty alcohol engines

Sponsor	Data Year	Model	Power (hp)	Fuel[1]	Ignition[2]	Catalyst[3]	Test[4]	Emissions (g/bhp-hr) PM	NO_x	CO	HC (OMHCE)	HCHO
Detroit Diesel	1992	DDC 6V-92TA	253	M100	GP	Ox	FTP	0.03	1.7	2	0.1	0.07
Detroit Diesel	1992	DDC 6V-92TA	253	M85	GP	Ox	FTP	0.03	4.1	1.6	0.2	0.08
Detroit Diesel	1991	DDC 6V-92TA	277	M100	GP	Ox	FTP	0.03	2.0	1.3	0.3	0.06
HD Truck[5]	1991	DDC 6V-92TA	300	M85	GP	None	FTP	0.09	3.4	6.7	2.2	0.12
HD Truck	1991	DDC 6V-92TA	300	M85	GP	Ox	FTP	—	3.4	1.1	0.6	0.07
HD Truck	1991	DDC 6V-92TA	300	M100	GP	None	FTP	0.11	2.9	3.2	2.2	0.12
HD Truck	1991	DDC 6V-92TA	300	M100	GP	Ox	FTP	—	2.6	0.3	0.2	0.06
HD Truck	1991	DDC 6V-92TA	350	M100	GP	None	FTP	0.2	2.6	3.8	2.0	0.14
MASSCAR[6]	1991	DDC 6V-92TA	286	M100	Avocet	Ox	8 M	0.05	4.75	0.64	0.58	—
Detroit Diesel	1992	DDC 6V-92TA	253	M100	Avocet	Ox	FTP	0.039	3.96	0.61	0.22	—
SwRI	1990	DDC 6V-92TA	249	M100	Avocet	Ox	FTP	0.12	2.69	1.72	1.28	0.07
HD Truck	1991	DDC 6L-71TA	300	M100	GP	None	FTP	0.12	3.3	3.5	1.8	0.16
HD Truck	1991	DDC 6L-71TA	300	M100	GP	Ox	FTP	0.02	2.7	0.5	0.1	0.05
MILE[7]	1986	Caterpillar 3406B	350	M100	GP	None	13M	—	2.65	4.71	2.38	0.07
MILE	1987	Caterpillar 3406B	350	M100	GP	None	FTP	0.15	3.05	12.4	4.45	0.49
MILE	1989	Cummins L10	270	M100	Avocet	None	13 M	—	4.78	1.5	2.8	0.13
HD Truck	1991	Cummins L10	240	M100	Avocet	Ox	8 M	0.018	6.01	0.063	0.026	—
HD Truck	1989	Navistar DT-466	210	M100	GP	None	FTP	0.12	4.7	3.4	1.06	—
HD Truck	1990	Navistar DT-466	210	M100	GP	Ox	FTP	0.05	3.52	0.06	0.06	—
Daimler Benz	1987	OM 4070	—	M100	Avocet	None	—	0.08	7.0	—	1.3	—
CEC[8]	1985	MAN D2566 UH	200	M100	Spark	Ox	13M	—	5.75	0.28	0.5	—
HD Truck	1991	Ford 6.6L	170	M100	Spark	TWC	Otto FTP	—	1.21	21.1	0.32	0.011
Detroit Diesel	1992	DDC 6V-92TA	253	E95	GP	Ox	FTP	0.04	4.2	1.7	0.7	0.02
SwRI	1990	DDC 6V-92TA	249	E85	GP	Ox	FTP	0.41	4.49	8.65	2.44	0.12
SwRI	1990	DDC 6V-92TA	249	E160	Avocet	Ox	FTP	0.13	2.37	7.92	2.29	0.09
SwRI	1990	DDC 6V-92TA	249	E160	GP	Ox	FTP	0.11	1.44	10.02	3.46	0.13
Saab-Scania	1991	Scania DSI11E	260	E95	Avocet	Ox	ECE R49	0.04	3.3	0.07	0.15	—

[1]M100 = 100 percent methanol; M85 = 85% methanol and 15% gasoline; E95 = 95% ethanol and 5% gasoline, or 95% ethanol, 3% gasoline, and 2% Avocet, E85 = 85% ethanol and 15% gasoline; and E160 = 80% ethanol and 20% distilled water.
[2]GP = glow plugs; Avocet = fuel mixed with a percentage of Avocet ignition improver; Spark = spark plugs.
[3]Ox = oxidation catalyst, TWC = three-way catalyst; None = no catalyst.
[4]FTP = Transient Federal Test Procedure; 8 M = ARB 8-mode test; 13 M = 13-mode test; Otto FTP = Otto cycle Federal Test Procedure; ECE R49 = European HD Emissions test.
[5]Data from California Methanol-Fueled Heavy-Duty Truck Demonstration sponsored by the California Energy Commission and the South Coast Air Quality Management District.
[6]Data from the MASSCAR (Methanol and Avocet Supporting Southern California Air Resources) Project, sponsored by the South Coast Air Quality Management District.
[7]Data from the MILE (Methanol in Large Engines) Project, sponsored by Energy, Mines and Resources Canada.
[8]Data from the Methanol-Fueled Transit Bus Demonstration, sponsored by the California Energy Commission.

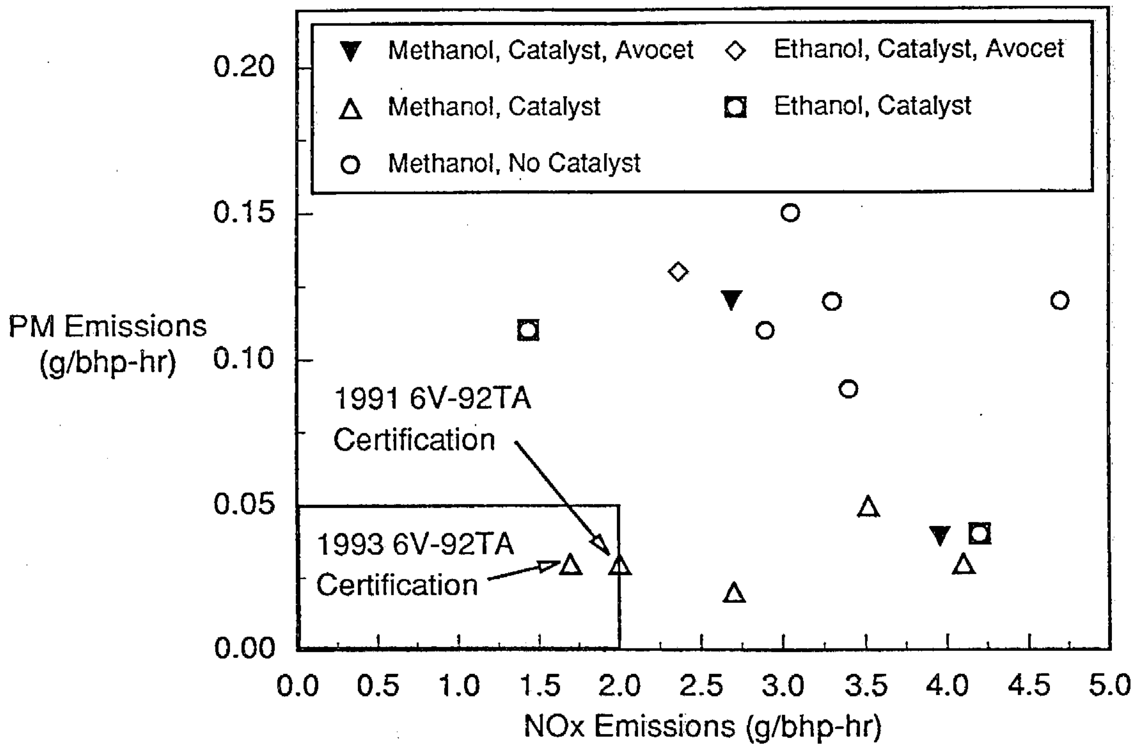


Figure 5-1. NO_x and particulate emissions from alcohol engines

homogeneous charge engines also show some promise, but have also seen very little development work.

DDC has put forth the most advanced methanol engine with their certified 6V-92TA engine. The certified engine meets the Scenario 1 goals. DDC's current development efforts are concentrating on increased oil control and retarded injection timing. Demonstrations have shown that the methanol version's diesel equivalent fuel economy is about 10 percent worse than that of the diesel version. Durability and reliability are approaching that of the diesel version.

Four stroke engines have not seen as much development effort as two stroke engines. Navistar has shown promising low emissions with their DT-466 engine. This engine will require more modification to take advantage of methanol's beneficial properties as well as to accommodate methanol's shortcomings. Navistar's development approach will use EGR, retarded injection timing, and improved oil control to reduce any further particulate emissions. Caterpillar is in a similar position to Navistar in that they are in the early stages of development and demonstration. The Caterpillar engines need similar measures to those noted above. In addition, a catalyst must be added to reduce methanol and formaldehyde emissions. Both Caterpillar and Navistar have put forth a great effort to support their engines running in demonstrations, indicating that the companies are serious about further developing their methanol programs. Cummins has turned its attentions towards CNG and has ceased development of the methanol versions of their engines.

Ignition improvers have shown themselves to be a viable alternative to glow plugs and spark plugs, especially important for retrofitting diesel engines with alcohols. Emissions tests have shown that methanol with Avocet can perform in the same realm of emissions as methanol with glow plugs, even on an engine optimized for glow plug use. Further development work should concentrate on optimizing for Avocet use, through changes such as timing adjustments. An example of a four-stroke, homogeneous-charge engine that has been developed for methanol is the Ford 6.6L. This engine is relatively clean compared to the heavy-duty Otto-cycle emissions limits. Since it operates with a three-way catalyst, low NO_x emissions make it a candidate for meeting Scenario 2 emissions, but the commercial status of this engine is uncertain.

Methanol engines show great potential in meeting the Scenario 1 goals using current techniques of emission control. Scenario 2 emission levels will be seen by several methanol engines as research and development efforts continue. With increased engine oil control, methanol should be able to meet diminishing PM standards. Further NO_x reductions will require the use of advanced technologies that are applicable to diesel engines. These technologies include EGR and DE-NO_x catalysts. Particulate emissions do not increase with methanol engines when NO_x emissions are reduced; although there is a tradeoff of NO_x against CO and HCs.

SECTION 6

GASEOUS FUELS TECHNOLOGIES

Gaseous fuels are another alternative fuel option for heavy-duty engines. Over 30 demonstration programs throughout the country currently use gaseous fuels. Gaseous fuels most typically used in heavy-duty engines are natural gas (NG), either in a compressed or liquefied form, and liquid petroleum gas (LPG). Most of the technology discussed here will work with either NG or LPG, however LPG requires some changes in engine design relative to NG. The following sections will consider both fuels together, pointing out differences where they apply.

6.1 FUEL PROPERTIES

Natural gas is primarily methane, while LPG is a by-product of the refining process of crude oil and natural gas. Both NG and LPG have high hydrogen to carbon ratios and produce less CO_2, a principal greenhouse gas. However, methane is also a greenhouse gas, and is growing at a rate similar to CO_2. NG reacts to form very little ozone. Methane is practically inert in the formation of photochemical smog. LPG and its combustion products are more reactive, but still much less reactive than gasoline and its products of combustion. A description of the two fuels is found below. Table 6-1 lists the properties of compressed natural gas (CNG), liquefied natural gas (LNG) and LPG.

6.1.1 Liquefied Petroleum Gas

LPG is a liquid fuel when stored under pressure but is typically introduced into the engine cylinder as a gas. LPG consists primarily of propane with up to 10 percent propylene (propene), but sometimes can contain butane and butylene. LPG engine fueling technology is similar, but not identical to NG. As indicated in Table 6-1, the flame speed, octane number, and energy density are

Table 6-1. Properties of CNG, LNG, and LPG

Properties	Natural Gas		LPG	Diesel Fuel #2
	CNG	LNG		
Molecular weight	18.06	16.74	44.25	≈170
Element (wt %) C H	73.90 26.10	74.69 25.31	82.13 17.87	86.88 13.12
Density (@ 60°F, 1 atm) (lb/ft^3) (lb/gal)	10.5[1] 1.4	25.4[2] 3.4	31.58 4.221	52 to 55 7.0 to 7.3
Boiling point (°F @ 1 atm)	-258.7	-258.7	-43.73	325 to 750
Freezing point (°F)	-296.5	-296.5	-305.8	< 20
Vapor pressure (psi @ 100°F)	—	—	188.0	negligible
Heat of vaporization (Btu/lb)	219.2	219.2	183.0	116
Net heating value (Btu/lb) (Btu/gal)	20,172 28,241	21,235 72,199	19,898 83,989	18,100 128,000
Stoichiometric mixture net heating value (Btu/lb)	1,197	1,247	1,285	1,248
Autoignition temperature (°F)	1,170	1,170	842	494
Adiabatic flame temperature (°F)	3,549	3,551	3,637	3,620
Flame speed @ stoichiometry (ft/s)	1.2	1.2	1.3	—
Octane –Research –Motor	>120 >120	>120 >120	112 97	— —
Cetane	—	—	—	> 40
Flammability limits (vol% in air)	5.0 to 15.0	5.0 to 15.0	2.1 to 9.5	1.0 to 5.0
Stoichiometric air/fuel mass ratio	16.85	17.03	15.49	14.5
Stoichiometric air/fuel volumetric ratio	9.22	9.42	23.55	85
Sulfur content (wt%)	0	0	0	< 0.05
Approximate relative fuel volume for equivalent range	4.3	1.7	1.5	1
Approximate relative full fuel tank weight for equivalent range	3.7	1.4	1.2	1

[1]CNG density at 3,000 psi and 60°F.
[2]LNG density at 20 psi and -260°F.

different for NG and LPG. LPG has a lower octane rating than NG, thus has a lower knock-limiting compression ratio. This hampers its use in the heavy-duty engine area. High propylene content of the fuel can further reduce knock limited compression ratios and increase the reactivity of the fuel.

As LPG is petroleum based, its cost and availability tracks that of oil, and so LPG is not the most attractive alternative fuel from an energy independence perspective. Nevertheless, current estimates suggest that there is enough LPG to cost-effectively fuel several million vehicles in California.

6.1.2 Natural Gas

Natural gas is typically composed of 85 to 99 percent methane; the remaining composition includes nitrogen, helium, carbon dioxide, hydrogen sulfide, ethane, propane, butane and other trace constituents. Conventional industry opinion is that NG hydrocarbon exhaust emissions are very closely related to the hydrocarbon composition of the fuel. Combustion of methane does not usually result in higher hydrocarbons. Formaldehyde production from lean burn NG engines is usually very low and will only approach methanol levels with poor combustion system characteristics.

OEM engines will benefit from integrated electronic control as with gasoline and diesel engines. The quality and reliability of retrofit conversions is a potential concern from many viewpoints, NO_x and other emissions will remain a concern unless legislated. There are two approaches for addressing the NO_x issue: tightly-controlled stoichiometric combustion with a NG-compatible three-way catalyst, and lean-burn combustion with or without an oxidizing catalyst. For heavy-duty applications where CNG might supplant diesel engines, CNG offers the additional advantage of negligible particulate emissions. This is important relative to the California 1991 bus and 1994 truck particulate standards.

NG has a higher energy density (mass basis) than either gasoline or diesel fuel. Additionally, its high octane rating can utilize higher compression ratios than gasoline engines which results in better fuel economy, and its wide flammability limits allow better NO_x control through lean operation. However, NG's low cetane characteristics disallow using it in conventional compression-ignition systems. The properties of LNG may provide an advantage over CNG in some engine applications. Since LNG is stored at -260°F, it can contain fewer impurities than CNG depending on the amount of purification. CNG components such as propane, butane, CO_2, and low levels of moisture are either liquids or solids at -260°F. Therefore, these components are removed during processing and are not present in LNG. As a result of processing, LNG is a more consistent fuel than CNG and is low in those components which promote knock and change stoichiometric air/fuel ratio.

Any potential variation in the composition of NG could influence NG's heating value, hydrogen-to-carbon ratio, stoichiometric air/fuel ratio (A/F), specific heat, and other properties which may adversely affect the performance of natural gas vehicles (NGVs). Variations in NG composition can reduce octane rating, possibly leading to detonation and potential engine damage/failure. Furthermore, increased concentrations of higher hydrocarbons will affect lean flammability limits, energy content and photochemical reactivity of the fuel.

NG is stored in vehicles either as CNG or as LNG, necessitating different types of storage equipment. In either case, higher storage volumes and heavier fuel tanks for NGVs result in limited range when compared to current diesel and gasoline powered vehicles. CNG is stored in high pressure vessels between 3,000 to 3,600 psi. LNG must be stored at temperatures near -260°F in cryogenic vessels. The added difficulties of LNG are offset by the gains in storage and refueling times. As indicated in Table 6-1, LNG is volumetrically closer to diesel fuel than CNG, thus LNG provides increased vehicle range for a given tank volume or weight limit. Moreover, the actual refueling time is typically between 2.5 to 4 times faster for liquids than gases. While the storage and fuel delivery systems differ, the engine technology of CNG and LNG vehicles remains virtually the same. LNG storage tanks, however, must have pressure relief valves that vent to the atmosphere during extended periods of vehicle inactivity.

Unique to NG as an alternative fuel (and to some extent LPG) is the pre-existence of a distribution infrastructure which would ease the initial capital investment required (however, there is a large capital cost for compression equipment needed to increase the gas pressure to 3,000 psi). In addition, NG has been used in stationary HD engines for many years and some of the technology can be directly applied to mobile engines. These factors would aid in establishing it as a marketable fuel.

The density of NG is less than air and thus NG will dissipate quickly instead of lying on the ground in pools like gasoline or other liquids. Small amounts of LNG will quickly vaporize and act like NG. NG storage containers (both CNG and LNG) have been designed and constructed to rigid safety requirements for thermal cycling, fatigue, fire immersion and burst pressure. The high ignition temperature and quick dissipation properties of NG reduce the chance of ignition in an accident. CNG and LNG, however, can be significant hazards in closed spaces. Proper ventilation and removal of ignition sources near the ceiling of an enclosed area will reduce this hazard.

Overall, NG makes an excellent fuel for IC engines. Recent efforts by Caterpillar, Cummins, DDC, Hercules, Navistar, and others to produce commercially available HD vehicle engines attest to this fact. At the current pace of technological research and development, and given the results of current NG engines emissions testing, several NGVs will meet 1994 emissions standards and may lead to even more stringent requirements.

6.2 ENGINE TECHNOLOGIES

Several approaches to gaseous engines currently exist in either production or prototype form. All of the engines have been developed from either gasoline or diesel engines. A majority of these induct NG or LPG into the intake air and are usually throttled. These fall into a homogeneous charge category and require spark plugs to initiate ignition of the air/fuel charge. In addition, several manufacturers have developed pilot injection engines which inject diesel fuel into the NG/air mixture to provide ignition. Finally, DDC is developing a compression-ignition 2-stroke engine that injects NG into the cylinder near top dead center (TDC) at high pressure. These engine technologies are described below.

6.2.1 Homogeneous Charge Engines

Homogeneous charge engines are the most common type of gas engine since gaseous fuels are high octane and thus do not autoignite readily. Engines which operate on gaseous fuels are throttled and either carbureted, single point (throttle body) or multi point (port) injected. The simplest system utilizes a gas mixer which takes the place of a carburetor. There are two kinds of mixers: The venturi mixer, which operates with constant flow diameters using pressure differences to regulate flow, and the orifice meter, which operates at constant pressure and controls the volume of flow by varying the orifice diameter. Mechanical gas mixers require regulation of gas pressure relative to air pressure, non-sonic gas injectors are also sensitive to gas supply pressure. All CNG systems require regulation from the gas tanks to the low pressure regulators or gas injectors. Errors in air/fuel ratio from mechanical systems can result from sticking, hysteresis or wear (venturi mixers have no moving parts and as such do not wear). A well-engineered mechanical system can easily control gas temperature relative to air temperature. All systems, whether electronic or mechanical, are effected by gas composition changes.

Electronic fuel injection offers potential for lower emissions and improved control of the torque curve shape through more flexible control. Single point (throttle body) injection provides all the benefits of open or closed loop electronic control together with excellent gas/air mixing. Cylinder-to-cylinder air/fuel ratio variations are minimized with a well engineered single point (or carbureted) system. Potentially more expensive, multi point (port) injected systems provide a small additional benefit in terms of improved transient air/fuel ratio control by minimizing the volume of induction system containing the gas/air mixture. However, multi point injection systems currently offer poor gas/air mixing and cylinder-to-cylinder air/fuel ratio control. Performance of multi point injected heavy-duty engines currently only approaches that of single point injected systems if hand matched sets of injectors are used. The lowest emission levels demonstrated so far from heavy-duty engines come from NG engines that are single point injected stoichiometric systems.

A spark ignition system is needed with homogeneous engines to ignite the gas/air mixture. If a lean mixture is used and as compression pressure rises for a given bmep level, a stronger spark is necessary, requiring the use of a high energy ignition system. In most of the diesel engines that are converted to gaseous fuels, the diesel fuel injector is removed and replaced with a spark plug.

Two concepts in homogeneous engines are used to control emissions as shown in Figure 6-1, namely lean-burn, possibly with an oxidation catalyst, and stoichiometric with a three-way catalyst. A description of these engine technologies and representative engines is found below.

6.2.1.1 Lean Burn Engines

Lean-burn engines operate with excess air to reduce NO_x emissions during combustion. The lean air/fuel mixture requires a strong spark for combustion to be achieved. During combustion, the excess air absorbs some of the heat of combustion, reducing peak cylinder temperatures and thus NO_x emissions. However, HC emissions increase as the mixture becomes leaner and approaches the air/fuel ratio needed to control NO_x production, as shown in Figure 6-2. Partial oxidation of methane at lean conditions can produce formaldehyde. An oxidizing catalyst is sometimes used to reduce HC, formaldehyde and CO emissions. Methane, however, is difficult to control because it is a stable compound that oxidizes poorly at normal lean-burn exhaust temperatures. Lean-burn engines typically use turbocharging to compensate for the lower energy content of a given volume of lean gas/air mixture. The benefits of lean-burn engines are that they are durable and have high thermal efficiency, lower engine-out NO_x emissions, and a higher rating potential than stoichiometric engines due to improved knock resistance. The Cummins L10 is a leading heavy-duty natural gas engine and is the closest to large scale commercialization. Over 50 L10 engines are operating in transit buses in the United States and Canada. The natural gas version of the L10 is a lean-burn engine which produces 240 bhp at 2,100 rpm and 850 ft-lb torque at 1,300 rpm. The on-road fuel consumption is not as favorable for L10-equipped natural gas buses,

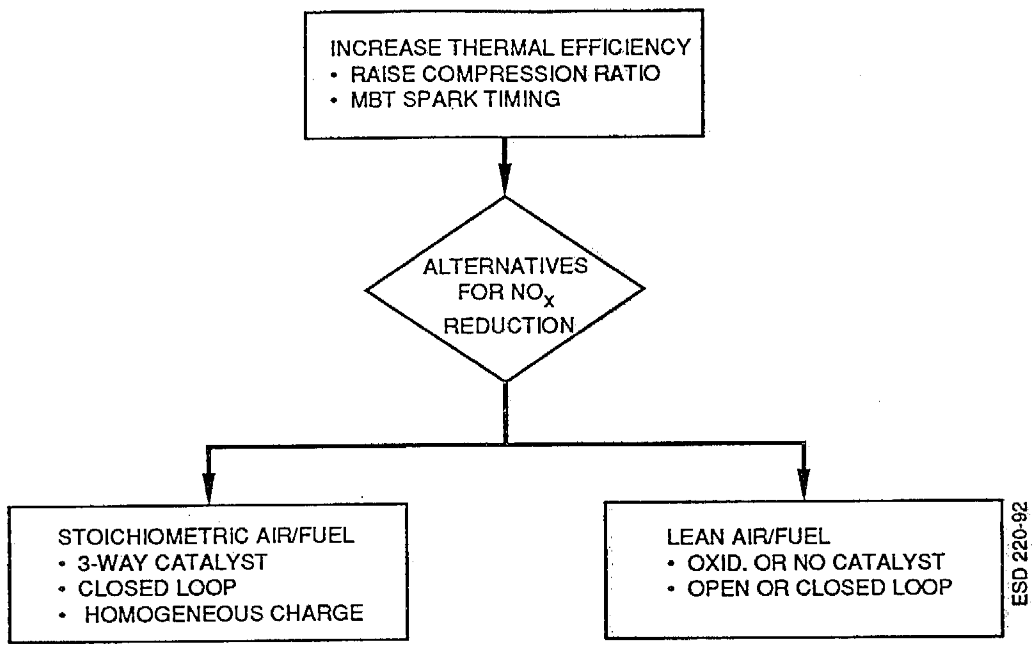


Figure 6-1. Homogeneous charge gaseous engine concepts for low NO_x emissions

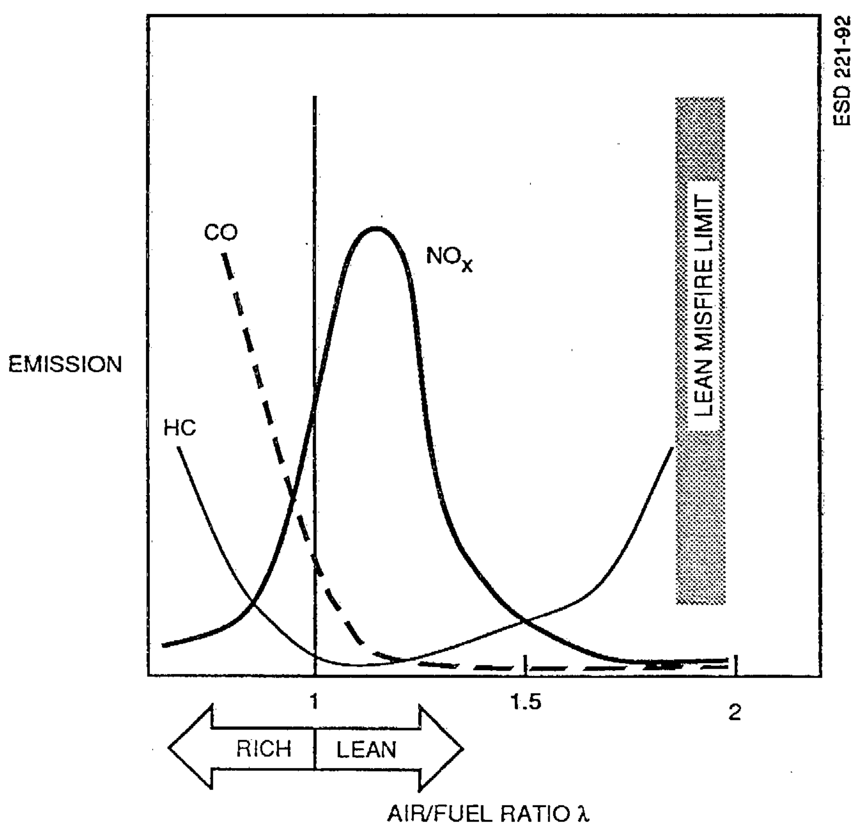


Figure 6-2. Exhaust emissions as a function of air/fuel ratio

which experience 25-percent worse diesel equivalent fuel economy compared with diesel buses. On-road or drive-cycle NG efficiency is compromised compared to diesel due to part load throttling. On higher speed duty cycles, this reduction in fuel economy would be less.

The Cummins L10 engine is equipped with a modified IMPCO natural gas mixer which is located downstream of the turbocharger. The engine is equipped with an Altronics electronic spark ignition system and a Woodward electronic throttle governor.

The air/fuel ratio must be properly adjusted to achieve good engine reliability. The engine is also equipped with a turbocharger waste-gate which also must be adjusted for proper engine power and reliability. Cummins is equipping their newest engines with a waste-gate controller. The air/fuel ratio varies with engine load. At idle, the engine operates at an air/fuel ratio of 20:1, which is relatively close to stoichiometric, and provides for stable combustion at low load. At peak torque and rated power, the engine operates leaner with air/fuel ratios ranging from 23:1 to 26:1.

Lean operation results in a trade-off between NO_x and HC emissions. Data from Cummins show that HC emissions, which are 85 percent methane, are 4.5 g/bhp-hr with NO_x levels of 3.5 g/bhp-hr. The HC emissions increase to 5.5 g/bhp-hr at 2.5 g/bhp-hr NO_x. Cummins plans to use a catalyst to lower THC, CO, and PM emissions. They recently certified the engine at 2.0 g/bhp-hr NO_x, 0.6 g/bhp-hr NMHC, 0.4 g/bhp-hr CO and 0.02 g/bhp-hr PM in August 1992. Cummins estimates that the price of this engine will be 2.3 times that of the current diesel engine due to the electronic governor, ignition system, catalyst, and warranty (Reference 49). A large portion of the additional cost is related to the warranty. Warranty costs are high primarily because it is a new product rather than a NG related issue. Cummins also plans to offer their B series 5.9 liter engine on natural gas in 1994.

Cummins is also demonstrating the L10 engine with LPG fuel at Orange County Transit District. The engine is equipped with a Deltec LPG mixer and fuel system. Currently, the engine operates in a lean configuration and is not optimized for LPG operation. Chassis dynamometer emission tests using the Central Business District (CBD) cycle were performed on buses equipped with this engine as well as the CNG engine in August 1992. Results for the LPG and CNG engines in comparison with the diesel version are shown in Table 6-2. The LPG engine is scheduled to be upgraded with improved fuel control and an oxidation catalyst.

The Caterpillar NG 3406 is also a lean-burn gas engine. It is turbocharged and air-to-air aftercooled. It is rated at 350 bhp at 1800 rpm with 36 percent efficiency at rated power and 38 percent efficiency at rated torque. The engine uses a Deltec venturi-type electronic mixer and an 11:1 compression ratio. This engine is currently being tested, but Caterpillar projects emission results of 2 g/bhp-hr NO_x, 0.1 to 0.15 g/bhp-hr PM, 2 g/bhp-hr CO and 3.5 g/bhp-hr THC (no more than 15 percent non-methane). An oxidation catalyst is used to control THC, CO and PM emissions (Reference 50).

Hercules has developed two lean-burn engines, the 3.7L and the 5.6L. The 3.7L engine produces 130 bhp at 2800 rpm and 300 ft-lbs torque at 1500 rpm. It uses open-loop electronically-controlled mixer, 10:1 compression ratio, turbocharging and aftercooling. Recent test results on this engine are 1.68 g/bhp-hr NO_x, 13.7 g/bhp-hr THC, 1.3 g/bhp-hr non-methane hydrocarbons (NMHC), 3.6 g/bhp-hr CO and 0.1 g/bhp-hr PM. The 5.6L engine is turbocharged

Table 6-2. Cummins L10 Engines on CBD cycle (gm/mi)

	CNG		LPG	Diesel
	Without Catalyst	With Catalyst		
NO_x	11.4	12.0	36.5	31.7
PM	0.05	0.03	0.04	1.23
CO	19.1	0.03	45.1	13.5
HC	18.6	2.60	10.5	1.24
HCHO[a]	1.30	0.01	1.35	0.08

[a] HCHO - Formaldehyde

and aftercooled with a 10:1 compression ratio. The engine is rated at 190 bhp at 2800 rpm with a peak torque of 450 ft-lb at 1500 rpm. Efficiency at wide-open throttle is approximately 35 percent. Recent test results for this engine are 1.56 g/bhp-hr NO_x, 3.97 g/bhp-hr THC, 0.76 g/bhp-hr NMHC, 2.09 g/bhp-hr CO and 0.06 g/bhp-hr PM (Reference 51). Hercules estimates that the 5.6L NG engine will cost 1.4 to 1.5 times as much as the 1991 certified, comparably-rated diesel engine (Reference 52).

SwRI has been developing their lean-burn NG Volvo engine under the Co-Nordic Natural Gas Bus program. The engine is a 6-cylinder engine with an open-chamber combustion system and gas mixer. It is turbocharged and aftercooled. The compression ratio is approximately 12:1 with a 9.6L displacement. SwRI had intended a rated power of 240 bhp at 2,000 rpm and peak torque of 900 ft-lbs at 1,500 rpm, however, they were unable to achieve the target torque curve and were forced to derate the engine by over 10% to contain NO_x emissions. The peak torque achieved was 665 ft-lbs at 1,400 rpm with a rated power of 228 bhp at 2,200 rpm. SwRI estimated FTP transient test emissions of 1.15 g/bhp-hr NO_x, 1.66 g/bhp-hr THC and 0.76 g/bhp-hr CO with the derated torque curve (Reference 53). The engine uses an oxidation catalyst to control THC and CO emissions. SwRI estimates that the incremental cost for this engine will be approximately $3,000 over the cost of the Volvo 9.6L diesel engine (Reference 54). Additional warranty costs will also be added.

Navistar and Ricardo Consulting Engineers have been working together to develop a lean burn CNG version of the 7.3L V8 diesel engine. This program is sponsored by Southern California Gas, NYSEG, Santa Barbara Air Pollution Control District and the Gas Research Institute. The 7.3L CNG engine uses an open chamber featuring the Ricardo "Nebula" high turbulence combustion bowl together with a wastegated turbocharger without aftercooling or catalyst. Initial prototype engines use an 11:1 compression ratio, an IMPCO mechanical gas mixer and Altronic CD ignition system. The CNG engine is intended as a "no compromise" replacement for the existing NGD V8 diesel engine and replaces the diesel torque curve with peak torque of 450 ft-lbs at 1,800 rpm and rated power of 210 bhp at 2,800 rpm. The project has initially targeted compliance with 1994 heavy duty emissions standards and very low levels in the future. FTP transient demonstrations were carried out at Ricardo with the following emissions: NO_x at 3.38 g/bhp-hr, THC at 5.14 g/bhp-hr, NMHC at 0.5 g/bhp-hr, CO at 2.84 g/bhp-hr and PM at 0.1 g/bhp-hr. Peak thermal efficiency has been measured at over 37 percent but more importantly, thermal efficiency over the FTP transient test cycle is only 5 to 10 percent reduced from the existing 7.3L IDI diesel engine. Several engines will be built for field demonstration in 1993.

6.2.1.2 Stoichiometric Engines

Stoichiometric engines operate near the chemically correct air/fuel ratio for oxygen free exhaust gas and use a three-way catalyst (TWC) to control emissions. For optimum efficiency of the TWC, these engines must be operated very close to the stoichiometric A/F to simultaneously oxidize THC and CO while reducing NO_x. The benefits of the stoichiometric engine with a TWC are extremely low emissions while still providing high power output, though detonation limits power output to lower levels than can be achieved with lean-burn engines. However, they operate at higher combustion and exhaust temperatures and are subject to knock limitations which may limit their durability. EGR, used in combination with a stoichiometric/TWC engine, will result in lower combustion temperatures to improve durability and reduce NO_x. Engine-out emissions of stoichiometric engines are very high, so failure of the TWC or fuel/air control would result in emission levels several times the 1991 standards.

Most of the current engines that use the stoichiometric/TWC system are converted from gasoline engines. General Motors 4.3L, 7L and 7.4L engines were converted by various researchers to natural gas. In addition, some diesel engines have also been converted to stoichiometric/TWC operation with gaseous fuels.

The first certified NG HD engine was the Tecodrive 7000 in 1991, a GM 427 cid 7L engine producing 214 bhp. The engine is equipped with an IMPCO natural gas mixer and operates with an open-loop control system. The engine uses EGR to further control NO_x emissions. With deterioration factors, this engine certified at 1.4 g/bhp-hr NO_x, 14.5 g/bhp-hr CO and 0.3 g/bhp-hr NMHC (Reference 55). The engine is currently being used by the school districts as part of the California Energy Commission (CEC) School Bus Demonstration Program where 10 buses are being used in normal service. CEC plans to add another 100 buses equipped with a turbocharged version of the engine in 1993. Houston Metro is also using this engine for three of their pick-up and delivery trucks fueled on LNG. The price of these engines is estimated to be $4,000 (Reference 56).

Brooklyn Union Gas (BUG) converted the GM 7.4L engine to natural gas, using the stoichiometric/TWC approach. This engine produced NO_x levels of 1.2 g/bhp-hr and 230 bhp. This engine, as well as other stoichiometric CNG engines, exhibited exhaust temperatures which were significantly higher than those found in a diesel engine. The engine installation would require attention to exhaust insulation. EGR would mitigate the high exhaust temperatures and further reduce NO_x (although the TWC is the primary control for NO_x).

Caterpillar converted their 3306 diesel engine to NG. This engine produces 250 bhp at 2100 rpm and 850 ft-lbs torque at 1200 rpm. This engine is an in-line 6 cylinder engine that is turbocharged and air-to-air aftercooled. The compression ratio for the natural gas version is 10.5:1 and the LPG version is 8:1 (with reduced power and torque). The engine uses a Deltec electronic venturi-style gas mixer and feedback control with an oxygen sensor and three-way catalyst. The target emission levels for this engine are 1 g/bhp-hr NO_x and 0.02 g/bhp-hr PM, but this engine is in the initial testing stage and needs refinements to meet those goals. First test results at Caterpillar's Peoria facility produced emission levels of 2.5 g/bhp-hr NO_x, 3 g/bhp-hr CO, 0.02 g/bhp-hr PM and 6 g/bhp-hr THC (over 90 percent methane). The efficiency of the engine at rated torque conditions is 37 percent, but drops to 33.5 percent at rated power. Caterpillar estimates the price of this engine to be approximately 2.5 times the cost of the comparable 1991 diesel engine (Reference 57), most of which relates to new product warranty costs.

Saab-Scania together with Ricardo Consulting Engineers have converted the Scania 11L engine to natural gas as part of the Co-Nordic Natural Gas Bus Project. The engine uses a wastegated turbocharger, air-to-air aftercooling, cooled EGR dilution, a single point injection system comprised of BKM Servojet injectors injecting via a Deltec venturi air/gas mixer and an advanced electronic control system. The engine produces 240 bhp at 2000 rpm and 738 ft-lb torque at 1,400 rpm. Measured FTP emissions for this engine on natural gas are 1.38 g/bhp-hr NO_x, 0.73 g/bhp-hr THC, 1.26 g/bhp-hr CO and 0.016 g/bhp-hr PM (Reference 58).

TNO of the Netherlands has been developing LPG engines for over a decade. The LPG fuel systems are manufactured by Deltec. Deltec converted over 100 M.A.N. buses with the G2866 engine to operate on LPG. This naturally aspirated engine produces 237 hp and operates with a TWC. These buses have been operating for over 5 years and NO_x emissions are 4.3 g/bhp-hr on the European test cycle. The engine efficiency is 36 percent. Since the engine operates stoichiometric, the on-road diesel equivalent fuel economy is about 30 percent worse than the diesel version. More recently, Deltec developed an Iveco 8469-21 engine for LPG operation. This engine is equipped with a Deltec electronic control system and has NO_x emissions below 1 g/bhp-hr. Deltec is also developing a 10.5 L engine for the U.S. market with similar emissions.

6.2.2 Diesel Pilot Injection Engines

Pilot injection is a bi-fuel concept to provide ignition of a NG/air mixture. As diesel fuel will autoignite readily, it provides an excellent ignition source for natural gas, which is very difficult to autoignite.

In these engines, natural gas is first inducted or injected into the cylinder at low pressure while the piston is near bottom dead center (BDC). The gas/air mixture is then compressed. Diesel fuel is injected into the cylinder near TDC. Upon injection, the diesel fuel autoignites and then ignites the surrounding NG/air mixture. A difficulty with this approach is optimizing the compression ratio such that diesel fuel will autoignite when injected, but the NG mixture will not knock or detonate. A micro-spray device is also being developed promoting even faster flame spreading and shorter combustion periods.

An example of this type of engine is the DDC "PING" (Pilot Injection Natural Gas) engine, which is a 6V-92TA coach engine converted to use natural gas and pilot injection. The PING engine uses a low pressure gas injector at the bottom of the stroke to inject natural gas directly into the cylinder. At full load conditions, the diesel pilot injection accounts for 15 percent of the fuel. The PING engine is turbocharged and aftercooled with a 15:1 compression ratio and DDC plans to achieve the same rated power and torque as their diesel 6V-92TA coach versions. Thermal efficiency is expected to be 37 percent to 38 percent at peak torque. Target emissions for this engine are 4.7 g/bhp-hr NO_x, 1.0 g/bhp-hr THC, 0.50 g/bhp-hr CO and 0.10 g/bhp-hr PM. As this engine uses from 15 percent to 100 percent diesel fuel depending on load, its high emission levels are a result of the diesel fraction. DDC has decided not to certify this engine in California due to its marginal NO_x improvement (Reference 59, 60).

6.2.3 Two-Stroke Direct-Injection Engines

Normally NG cannot be ignited without a spark or diesel pilot injection. However, in studying autoignition temperatures for a variety of fuels as shown in Table 6-3, DDC realized that the concept they developed for their direct-injection methanol engines discussed in Section 5.1 would also work for natural gas. DDC is currently developing a 6V-92TA engine that autoignites

Table 6-3. Required auto-ignition conditions (°F)

Fuel Type	Auto-Ignition Temperature	Cooling Effect	Necessary Cylinder Temperature
Diesel	600	30	630
Methanol	880	300	1,180
Natural Gas	1,170	10	1,180
Coal/Water	400 to 700	400	1,100 max

natural gas without a spark or pilot diesel injection. This two-stroke engine, rated at 253 and 277 bhp, replaces the current diesel fuel injector with a high pressure gas injector to inject natural gas directly into the cylinder near TDC. Since the relative autoignition temperature for NG is similar to methanol, the concept of using the bypass blower to control in-cylinder temperatures is expected to work in this engine as well. The natural gas mixes with compression-heated air and hot residual gases. This combination allows for autoignition of the charge upon injection into the cylinder without the need for any other ignition source except on start-up.

As this is a direct injection concept, the engine is designed with a 23:1 compression ratio, a turbocharger and an aftercooler. DDC has achieved the 253 bhp and peak torque ratings of the diesel 6V-92TA engine and is confident of achieving the 277 bhp rating as well. Thermal efficiency at peak torque is 39 percent, and except for a few problems with injectors, requiring modification, the engine has done well under 270 hours of durability testing. The emission goals for this engine at 2 g/bhp-hr NO_x, 1 g/bhp-hr THC, 0.5 g/bhp-hr CO and 0.05 g/bhp-hr PM. DDC plans to certify this engine in early 1994 (Reference 58, 59).

6.3 DEVELOPMENT AREAS

Most heavy-duty NG engine research and development areas derive from problems in converting diesel engines to spark-ignition engines. The resulting engine from this conversion has lower efficiency, higher heat rejection, throttling losses and higher exhaust temperatures. Current research in NG engines have identified areas of concern for the continuing development of NG technology. Included among these issues are controlling engine temperature, cylinder pressure, fuel injection pressure, and combustion duration. As work progresses on dedicated engines, the implementation of this research will have measurable impact on the performance of engines.

There are five main focuses of research and development. These are fuel systems, combustion systems, ignition systems, exhaust aftertreatment, and closed-loop feedback systems. Work in all areas have proven effective in increasing engine performance and decreasing emissions.